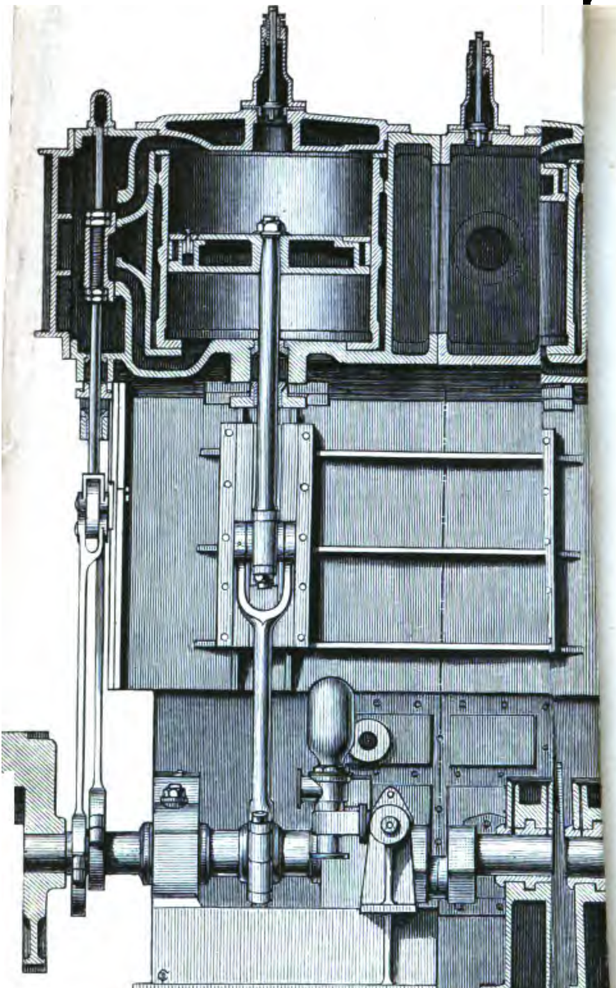
AN ELEMENTARY TREATISE

ON

STEAM.

SCHOOL CLASS BOOKS



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STEAM.

BY

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PREFACE.

THIS little treatise is mainly intended for the use of students who can solve simple equations in *Algebra*, and who know the simple definitions in *Trigonometry* and the simpler facts in *Physics*. In a few places the sign of summation is introduced, but there will be no difficulty in following the reasoning. It is recommended that readers follow up their mathematical studies, and that they make themselves thoroughly acquainted with such books as Dr. Everett's translation of Deschanel's Natural Philosophy: particularly Vols. I. and II.

As the subject of *Steam* has been treated in a manner very much like the present by other writers lately, I owe it to myself to state that this book has been in the hands of the publishers for more than two years.

I acknowledge the help afforded by the practical treatises of Bourne, Main and Brown, Clarke, and others, the Engineering papers, and above all by

the Manuals and scientific Papers of the late Professor Rankine. Professor Rankine worked successfully in almost every department of Engineering. His results are in many cases incomplete, and require experimental and other verification. Mainly by his labours, Mechanical Engineering is now to students what a country opened up in all directions by roads and railways is to settlers.

I acknowledge my obligations to the friends who have assisted me, and particularly to Dr. Everett, for his kindness in examining the proofs of the more important parts of the book.

JOHN PERRY.

CLIFTON, *July* 1873.

CONTENTS.

BOOK I.

HEAT, ETC.

BOOK II.

STEAM-ENGINES AND BOILERS.

BOOK III.

LOCOMOTIVES.

PAGE

CHAPTER IX.

CHAPTER X.

BOOK IV.

MARINE ENGINES.

CHAPTER I.

CHAPTER II.

CHAPTER III.

CHAPTER IV.

CHAPTER V.

CHAPTER VI.

BOOK I.

HEAT, ETC.

STEAM.

BOOK I.

HEAT, ETC.

INTRODUCTION.

1. We shall in general use English weights and measures in this book, but the common use of metres and grammes in scientific works renders the following short description of the *Decimal System* necessary:—

It would be well if there were one system only of weights and measures everywhere. Since we use the decimal system of numeration in arithmetic, it is natural to use weights and measures of which each unit of a higher denomination is equivalent to ten,, or some power of ten units of a lower denomination; so that we may work our calculations by decimals instead of by the old methods of reduction and vulgar fractions.

The *Metre* is the standard of length, being equivalent to 39·37079 inches. The *Gramme* is the standard of mass, being 15·432349 grains.

1 Millimetre	=	·03937079	inches.	1 Milligramme	=	·015432349	grs.
1 Centimetre	=	·3937079	,,	1 Centigramme	=	·15432349	,,
1 Decimetre	=	3·937079	,,	1 Decigramme	=	1·5432349	,,
1 Metre	=	39 37079	,,	1 Gramme	=	15·432349	,,
1 Decametre	=	393·7079	,,	1 Decagramme	=	154·32349	,,
1 Hectometre	=	3937·079	,,	1 Hectogramme	=	1543·2349	,,
1 Kilometre	=	39370·79	,,	1 Kilogramme	=	15432·349	,,

Cubic and square millimetres, &c., are measures of volume and area.

For the sake of convenience, the cubic decimetre is called a litre. It is rather less than our English quart.

To connect volume and weight, the unit of weight or gramme is the weight of a cubic centimetre of pure water at 4° C.

A kilogramme is equivalent to 2·2046 English pounds avoirdupois.

2. The pressure of one atmosphere, or 29·92 inches of mercury, or 760 millimetres of mercury, is 1·033 kilogramme per square centimetre, or 14·73 lb. per square inch. This is often roughly given as 15 lb. The weight of a cubic foot of dry air at 0° C., and at the pressure of one atmosphere, is ·0807 lb.

The weight of a cubic foot of pure water at 4° C. is 62·4 lb.

Motion is change of place. Recurring motions, such as those of the planets round the sun, give the best measure of time. The duration of the mean solar day is the absolute unit of time. The $\frac{1}{86400}$th of this, or *the Second*, is a unit of time more commonly employed by scientific men. A simple pendulum 39·13983 inches long has one second for its time of vibration at London. Time is as capable of measurement as length.

The ordinary beam and scales measure the relative *Masses* of bodies. The mass of the same body is the same everywhere in the universe, but the *Weight* of a body, or its tendency to move downwards, varies with its position with regard to the earth. This variation is usually neglected in practical treatises, and the weight of 1 lb. at any part of the earth is taken as the unit of weight or of force generally. In treatises on Mechanics we find that the mass of a body is given as its weight divided by 32·2, which is a number

approximately connecting mass and weight at London, where a foot is used as the unit of length. Strictly speaking, the *mass* of 1 lb. ought to be the unit of mass, and hence (see treatises on Mechanics) the absolute weight of the mass of 1 lb. at London is 32·2, and at other places is more or less than 32·2, according as gravity is more or less powerful than at London. This more scientific system of measurement will not be followed by us.

The *Density* of a body is its mass divided by its volume.

The *Specific Gravity* of a body is its density as compared with that of water.

Motion is change of place; and *Velocity*, when uniform, is the space passed over by a body as compared with the time.

Force is anything which alters the motion of a body; the amount of alteration during every second being proportional to the force.

The amount of motion of a body is measured by its velocity and its mass, and may be called *Momentum:* hence the momentum given to a body in a unit of time measures the force. The weight of 1 lb. will be employed by us as the unit of force.

Force has to be communicated from one body to another. The communicator is subjected to *pressures* or *tensions*, the former name being given to pushing forces, and the latter to pulling forces.

Force multiplied by the distance through which it moves a body is a measure of the work done on the body by the force.

When a fluid expands, the work done is measured by the increase of volume of the fluid × the mean pressure during expansion.

Elasticity (Art. 17) is the relation of the increase of pressure applied to a body to the compression produced.

Friction between two surfaces is independent of the area, depends on the nature of the surfaces, is proportional to the pressure keeping them together, and is independent of the relative velocity of the surfaces, unless the velocity be very great. The *Co-efficient of Friction* is a certain fraction which, when multiplied by the normal pressure, gives the friction.

Centrifugal force acts in the direction of the radius of the circle in which a body is moving, and as usually measured in pounds is $m\frac{v^2}{r}$, where $m = \frac{\text{weight}}{32\cdot 2}$, where v is the velocity in feet per second, and r is the radius of the circle in feet.

3. Physically, a substance may appear under three aspects, in which its particles are in three different states of aggregation: the solid, liquid, and gaseous states of matter.

Solids resist changes of volume and of form; liquids resist changes of volume but not of form,* and gases are always tending to increase their volume, filling the closed vessels in which they are contained.

As a general rule, when sufficient heat is given to a solid it becomes a liquid; when sufficient heat is given to a liquid, it becomes a gas. The temperatures at which changes of state occur are in general the same when the external conditions of pressure, &c., are the same. Increased pressure alone, without decrease of temperature, tends to solidify liquids † and to liquefy gases. Carbonic acid and other gases have been liquefied by the simultaneous use of freezing mixtures and increased pressures.

Water is an aggregation of molecules, each formed by the combination of two atoms of hydrogen with one of

* Liquids resist stresses, producing changes of form when they are suddenly applied; this property of liquids is called viscosity.

† Liquids which shrink in solidifying; the reverse is the case with water and melted cast iron.

oxygen. We imagine that with delicate and powerful instruments we might get a small portion, a particle of water, incapable of *physical* sub-division; this would be a molecule, and would be capable of *chemical* sub-division into its atomic constituents.

The molecules of water slide past each other more readily than the molecules of ice, and the molecules of steam are understood to be continually darting about through space, striking each other at intervals; but in all cases a molecule of ice, water, or steam consists of two atoms of hydrogen and one atom of oxygen chemically united. In fact, all bodies are composed of molecules incapable of further physical sub-division, each of these being a little system of atoms.

Although the molecules of a body require much force to separate them, we may suppose that they oscillate about certain positions as a pendulum oscillates about a vertical line. This supposition agrees with certain facts which will be discussed when we come to consider *Energy*, and which show that *Heat* is a form of energy, and justify us in saying that the heat of a body consists of oscillatory motions of its molecules. When these motions cease, there is a total absence of internal energy or heat.

We cannot expect to render any body absolutely heatless, for all bodies are, to a certain extent, conductors, and radiation is always going on from surrounding matter; but, as we shall shortly see, we are justified in hypothetizing an absolute zero as a basis to our readings on the thermometer.

Increase of temperature is increased rapidity of vibration. For all bodies there is a certain instant at which the increased length of swing is so great, that from the lessening of the forces of attraction, or from their being balanced by momentum, the molecules are enabled to move easily about each other, and the substance liquefies. We might overcome the forces of attraction and cause the particles to slide past each other by a proper application of external forces. This fluidity is shown in the metals by their malleability and ductility. Steel may be drawn out as a tube, or as a wire, even when quite cold.

According to the molecular theory of matter, of which I have been attempting to give a short description, a molecule

of a liquid slides past its neighbours in such a way that it is always very little separated from other molecules; or, as sometimes stated, it is always in contact with other molecules, and at the end of a certain period of time has been in contact with every individual molecule in the same vessel. An indication of this motion may be obtained by allowing a few drops of coloured water to fall into a vessel of clear water; in time, the coloured particles are to be found in every part of the vessel.

The molecular velocities get greater as the temperature rises, until the gaseous state is reached, in which the little masses are darting about in all directions, coming into contact with each other and with the sides of the containing vessel, and so changing their velocity both in direction and in amount, all the time retaining internal vibrations which continuously alter in length of swing, but never alter in period.

Pressure of a gas against the sides of a containing vessel is caused by the incessant impact of molecules.

The laws of Boyle, Gay-Lussac, and Charles (Art. 21), the phenomena of evaporation and condensation, of radiation and absorption of heat, may be mentioned as important deductions from the molecular theory as developed by Professor Clerk Maxwell.

CHAPTER I.

TEMPERATURE.

4. WHEN the contact of two bodies has no tendency to alter the heat-conditions of either, they are said to have the same temperature. The temperature of a body is its state with regard to sensible heat. We can say that some bodies are hot, and others warm, or cool, or cold; and we judge of the state of a body by its influence on our hands, or on matter in general. Now, when we say that one body is warmer than another, we have compared the influences of the two bodies on our hands as we touched them, or on other matter, and we have found that the *effects* differed only in degree, and we have judged them by a common standard, or by one another as standards; hence, when these effects produced by the hotness of a body may be measured and tabulated,* different numbers or marks on this table will indicate the effects producible by different temperatures. These numbers get connected with the temperatures at which the measured effects were produced; but they will have no meaning unless accompanied by information as to the substance affected, and the conditions under which the table was made out.

For purposes of measurement we must select some definite effect produced by heat; for example, the ex-

* The hand is defective as a thermometer, because there are many different effects produced by touching a heated body, and there are no means of measuring them.

pansion of some substance which expands in a regular manner when heated.

Bodies in general expand when their temperature is increased. A brass ball when heated will no longer pass through the ring which it just passed through when cold. Mercury which filled a glass vessel when cold, overflows when heated, because mercury expands more than glass. A closed bladder partially filled with cold air becomes well filled—sometimes to bursting—when held before the fire.

There are exceptions to this law. Water contracts as its temperature changes from $0°$ C. to $4°$ C., this latter temperature being the point of maximum density of water.

5. Taking a cubic foot of each of the gases, hydrogen, oxygen, air, nitrogen, nitric oxide, carbonic acid, and sulphurous acid, when at the same temperature as melting ice, or when surrounded by melting ice, and subjecting the gases to the same pressures at the same temperatures, we shall find that their volumes are always very nearly equal; that is, the hydrogen, oxygen, common air, nitrogen, and nitric oxide (gases which have never been liquefied by either great cold or excessive pressure) have volumes very nearly identical; but the condensible gases, carbonic acid and sulphurous acid, have volumes which differ from the rest and from one another. We say that the incondensible gases approximate to a *perfect gas*, and we use the expansion of this perfect gas when under constant pressure for our tabulation of temperature. What are called absolute temperatures are measured in this way.

Temperatures at which there are regular changes in volume are tabulated in numbers, which have a constant difference. Since this gas is indefinitely removed in temperature from its point of liquefaction, we may suppose that an increase in its volume from v to $v+m$, indicates the same change in state as from V to $V+m$. It is only under this supposition that we may *compare* differences of temperature.

Further on, we shall consider a scale of temperature con-

structed under a different supposition from that just given, the effects produced by unit rise in temperature being capable of connection with the fundamental units of mass, length, and time. The scale of absolute temperatures to which I refer is that deduced from the supposed expansion of a perfect gas, and is known to differ slightly from the scale deduced from experiments on the expansion of common air.

It is found that within considerable limits the ratio of increment in volume to increment in absolute temperature is practically constant in the case of mercury; and since mercury is liquid at all ordinary temperatures, the mercurial thermometer is that most commonly used for determining the temperature of a body. The volumes of a quantity of mercury, at the temperatures of melting ice and boiling water, differ by an amount the hundredth part of which indicates the expansion for a difference in temperature of one degree on this scale (Centigrade). In mercurial thermometers required for accurate work, the readings on the scale are made to correspond with those of the air thermometer, so that the expansion of mercury is not the same for every degree; the differences are so slight, that they are commonly neglected.

Other effects of heat on bodies, as the production of a current of electricity when the junction of two metals is heated, are occasionally employed for observation of temperature; but the mercurial thermometer is much more convenient for most purposes.

The ordinary mercurial thermometer consists of a small glass bulb at the end of a fine tube, containing mercury; the mercury has room to expand in the direction of the closed end of the tube. This tube having nearly the same cross-section everywhere,* equal increments in volume of the mercury will be shown by equal lengths of the tube; so that if we fix on the stem two points, at which the end of the mercury column is found at the temperatures of melting ice and boiling water, the hundredth part of the length between the

* Tubes never have the same cross-section everywhere; they are "calibrated" by passing a short thread of mercury along them, the length of this thread at different places giving determinations of the variations in the cross-sections.

two points represents a (Centigrade) degree. As the glass also expands, making the bulb and tube larger, the indications of the thermometer at intermediate temperatures will to some extent depend on the quality of the glass employed. Graduation by comparison with an air thermometer prevents the occurrence of such discrepancies.

That a thermometer may be used depends on the fact that a hot and cold body placed in contact tend to become equal in temperature, the time required for this equalization depending on the nature of the bodies and of the contact. Heat is given from the hot body to the cold one in such a case; that is, the amount of heat given out by the hot body is equal to that received by the cold body. Thus we consider *heat* to be something the loss or gain of which is denoted (in general) by a fall or rise in temperature. Whatever heat may be, we may consider that to raise the temperature of 1 lb. of water from 0° to 1° needs only half as much heat as to raise 2 lb. of water from 0° to 1°; and hence heat is measurable, and is capable of undergoing mathematical processes, although, as was seen in Art. 3, heat is not a substance (23).

Energy is capacity to do work, and, according to our theory of the constitution of matter, heat separates the molecules of bodies; in fact, heat does molecular work, and is a form of energy, and may be *measured* like all other forms of energy.

6. **The Mercurial Thermometer.**—For an ordinary mercurial thermometer, a bulb is blown on a very fine tube; by a process for which the reader is referred to treatises on Heat, the bulb and tube are filled with mercury at a temperature somewhat greater than that of boiling water, and the end of the tube is hermetically sealed. As the thermometer cools, the mercury contracts more than the glass, the end of the column being at different points of the tube at different temperatures.

The instrument is surrounded with melting ice, and a mark is scratched on the stem to show the position of the end of the column at the temperature of melting ice. A vessel of peculiar construction enables us to

surround the bulb and stem with steam from water boiling under ordinary atmospheric pressure, and we scratch another mark on our stem. If the tube is of the same size everywhere, we divide the distance between the marks into one hundred equal parts, and write opposite the divisions names beginning with 0° C. and terminating with 100° C. Our scale is called the Centigrade scale.

7. If we had divided the above space into 180 equal parts, and given the divisions names, beginning with 32° F. and ending with 212° F., we should have adopted Fahrenheit's system of graduation.

To change the reading of a temperature from one scale to another, we adopt the formula $F. = \frac{9 C.}{5} + 32°$, where F. and C. represent the readings on the Fahrenheit and Centigrade scale respectively.

For other methods of graduation, for corrections to be applied to the readings on ordinary thermometers, and for other thermometers than the one here described, students are directed to treatises on Heat.

8. **Changing Readings.**—*Example:* What on the Centigrade scale will indicate the temperature of 47° F.? Now, 47° F. means 15 of Fahrenheit's degrees above the melting-point of ice. Find the number of degrees on the C. scale which corresponds to 15 on the F. scale by the proportion 180 : 100 :: 15 : 8·3 the answer; that is, 47° F. is the same as 8·3 of Centigrade degrees above the freezing-point, or 8°·3 C.

Example.—What on the F. scale will indicate the temperature 89° C? Now, this temperature is 89 of the C. degrees above the temperature of the melting-point of ice. Find how many of the F. degrees will correspond to this by the proportion 100 : 180 :: 89 :160·2. To this 160·2 of F. degrees above the freezing-point add 32, and we get 192°·2 F. as the required reading.

Exercises.

(1.) What is the C. equivalent to a difference of temperature of 15 of the F. degrees?—*Ans.* 8°·33.

(2.) The table in Art. 9 is given according to the C. scale: what are the corresponding readings on the F. scale?—*Ans.* 220° F., 38°·9, 32°, 39°·2, 97°·88, 212°, 968°·8, 2786°.

(3.) Zinc boils at 1204° F., mercury at 608° F. Change these readings.—*Ans.* 650° C., and 320° C.

(4.) Change the following readings:—Polished steel is of a deep blue colour at 580° F. Polished steel is of a pale straw colour at 460° F. Sea-water freezes at 28° F.—*Ans.* 304°·5 C., 237°·75 C., −2°·2 C.

(5.) A unit of heat is the heat given to 1 lb. of water to raise its temperature 1° C.: how many units of heat are involved in raising 3 lb. of water through 30 of the F. degrees?—*Ans.* 50.

(6.) The latent heats of 1lb. of water and 1 lb. of steam are respectively 79 and 537 units, according to the C. scale: what are they according to the F. scale?—*Ans.* 142 and 966·6 units.

9. **Pyrometers.**—For roughly indicating high temperatures, engineers often use the expansion of a bar of platinum, allowing the free end of the bar to move the small arm of a lever working an index.

The only pyrometers which can be depended upon for accuracy, and whose readings can be connected with the ordinary scale, are those in which use is made of the expansion of gases and vapours.

It has been established that the further vapours are removed in temperature from the point at which they would condense to liquids, the more do their expansions approximate to that of a perfect gas or of common air.

The latest form of the air pyrometer is that constructed by Berthelot for temperatures between 300° and 500°. The iodine pyrometer and others give accurately the temperatures of furnaces according to the ordinary scales. Siemens has designed a pyrometer which depends on the principle that the electrical conductivities of platinum, iron, &c., decrease as the temperature is increased; these decrements may be used to indicate temperatures. A certain length of platinum wire in a voltaic circuit is raised to the high temperature, and measurements of the voltaic current give data for calculating the increased resistance of the wire, and thence the temperature.

Table of remarkable Temperatures.

Greatest artificial cold	— 140°
Mercury freezes	— 39°·4
Ice melts	0°
Greatest density of water	4°
Blood heat	36°·6
Water boils	100°
Red heat	526°
Cast iron melts	1530°

CHAPTER II.

EXPANSION BY HEAT.

10. We saw (3) that the particles of a body are always in a state of vibration, and that, in general, increments of heat increase the amount of this vibration, and thus push the particles further asunder. We have no means of observing the individual particles, but the accumulation of the separations is quite perceptible in the whole bulk of the body.

11. Solids expand by heat, get longer, broader, and thicker. Liquids expand by heat, increasing in volume. Gases tend to expand by heat; and when they are contained in closed vessels, this tendency to expansion shows itself in increased pressure on the inside surface of the vessel.

12. **Expansion of Solids.**—In isotropic solids, such as glass, metals, and other substances which are neither fibrous nor crystalline, where the molecules are alike in every respect, the body must expand at an equal rate in all directions. For instance, in a rectangular body $6'' \times 4'' \times 2''$, if the shorter side lengthens by a distance a, the $4''$ side will lengthen by $2a$, and the longest side by $3a$.

It has been found that if a body expands by a length a, when raised in temperature one degree, the approximate expansion for six degrees of increase in temperature will be $6a$, and in fact a for every degree of rise in temperature within moderate limits.

13. We are now in a position to discuss what is meant by *the co-efficient of linear expansion.* If a rod of copper at 0° C. expands by the fraction ·00001717 of its length in rising to 1°, it will expand by a fraction of its length twice this, or ·00003434, in rising to 2°; it will expand by a fraction of its length three times this, or ·00005151, in rising to 3°; and in general, by a fraction of its length nearly t times this in rising to $t°$.

This ·00001717 is called the *co-efficient of linear expansion of copper.* It must be remembered that this is the co-efficient of linear expansion of copper in *all directions*, for increase in length, breadth, and thickness.

Co-efficients of Linear Expansion.

Glass	·00000861	Iron wire	·00001235
Platinum	·00000884	Copper	·00001717
Cast iron	·00001127	Brass	·00001894
Steel	·00001145	Lead	·00002818

14. It will now be plain, that when we multiply the length of a bar by the co-efficient of linear expansion, we get the amount of expansion of the bar for one degree of rise in temperature. Thus, if l is the length of the bar at 0°, and a is the co-efficient of expansion, $l\,a$ expresses the amount of expansion for a rise in temperature of one degree. For a rise of t degrees, the amount of expansion is expressed by

$$t\,l\,a.$$

The whole length of the bar at $t°$ is evidently $l + t\,l\,a$ or

$$l\,(1 + a\,t)$$

Now, if we are told to calculate the expansion of a bar of copper during a rise in temperature from, say 30° C., given its length at 30°, we may first find the length of the bar at 0° before using the formula $t\,l\,a$ given above; but if we reflect that the length of the bar at 0° C. will be very little less than its length at 30°, we see that the product of the length at 0° by the small quantity a will differ but little from the product of the length at 30° by a, and so

we use the formula

$$t\,l\,a$$

for the amount of expansion of a bar when raised in temperature t degrees, the length at the lower temperature being l.

Example.—At 0° C. a cast-iron beam is 12 feet long: what is its length at 1,000° C., supposing the law of similar increments to hold for that high temperature? Here 1,000 × ·00001127 = ·01127, and this multiplied by 12 gives ·13524. Hence, 12·13524 feet is the answer.

Example.—At 25° C. a bar of wrought iron was 16 feet long. What is its length at 96° C.? Here the difference in temperature is 71 degrees, and 71 × 16 feet × ·00001235 or ·01403 is the amount of expansion, and 16·01403 feet is the answer.

In accurate work it is well to attend to the general equation. Length at t° = length at $0^\circ \times (1 + a\,t)$.

Example (exact method of working).—A bar of platinum at 250° is 3·265 metres in length. What is the length of the bar at 0°, and what is the length at 36°? Here the equation becomes 3·265 = length at $0^\circ \times (1 + 250 \times \cdot 00000884)$, from which

$$\text{length at } 0^\circ = \frac{3 \cdot 265}{1 \cdot 00221} = 3 \cdot 2578.\text{—}Ans.$$

Now, let us use this length at 0° to find the length at 36°. The equation becomes—

Length at $36^\circ = 3 \cdot 2578\,(1 + 36 \times \cdot 00000884) = 3 \cdot 259$. —*Ans.*

The same answers may be obtained by the less exact method of working sketched above.

Exercises.

(1.) By how much would the length of a submarine copper cable at 0° shorten, if the temperature became −20° C.?—*Ans.* ·0003434 of the length at 0°.

(2.) A wheel of wrought iron has an inside diameter of 5 feet when at the temperature of 970°. What is its diameter at 0°?—*Ans.* 4·9 feet.

(3.) A cylindric plug of copper just fits into a hole 4″ diameter in a piece of cast iron. After heating the mass to the temperature of 1,240°, by how much is the diameter of the hole too small for the plug?—*Ans.* ·0293 inches.

(4.) A bar of iron is 7 decimetres long at 0°. What is its length in boiling water? What is its length at 50°? What at 2,000°?—*Ans.* 7·007889, 7·003945, 7·15778.

(5.) Two rods, one of copper the other of iron, measure 9·8 decimetres each in length at 0°. What is their difference in length at 57°?—*Ans.* ·0027 decimetres.

(6.) What are the relative lengths of the brass and steel rods in a compensation (gridiron) pendulum,

where the expansion upwards of one set is equal to the expansion downwards of the other? If the whole length of brass rod is 7·9 decimetres, what is the whole length of steel rod?—*Ans.* The lengths are in the ratio of the co-efficients of expansion of brass and steel. The length of steel rod will be 13·07 decimetres.

(7.) Bars of wrought iron, each 3·4 metres long, are laid down at the temperature of 0°. What space is left between every two, if they are intended to close up completely at 90°?—*Ans.* ·0037791 metres.

(8.) The wooden pattern of a cast-iron beam must be longer than the casting at 0°. For a beam 12 metres long at 0°, what is the length of the pattern? Cast iron melts at 1,530°.—*Ans.* 12·207 metres.

15. It can readily be shown by Mathematics, that if a body increases its linear dimensions by a certain small fraction of them, the volume is increased by a fraction of itself almost exactly three times the original fraction. For instance, the side of a block of copper at 0° C. lengthens by the fraction ·00001717 of itself when its temperature becomes 1°; whereas the *volume* of the block increases by the fraction ·00005151 of itself. Thus ·00005151 is called the *co-efficient of cubical expansion of copper.*

16. It may happen that with rapid cooling the molecules of a solid do not return to their previous relative positions at the lower temperature.

This is the case in rapidly-cooled glass or steel, which only come back to their original state at a given temperature, when cooled slowly to that temperature. This slow cooling is called *annealing.*

17. That we may get some idea of the great stresses which act in a body when it is forcibly prevented from expanding, we shall consider the *Modulus of Elasticity.* A bar may be lengthened by forces

pulling it out; if the same forces be employed to push it in, the little decrement in length is exactly equal to the former increment. Thus, *exactly the same forces will lengthen or shorten a bar by the same small distance.*

It has been found that if a bar of iron lengthens or shortens by a distance a under the action of a certain statical force, a statical force twice as great will lengthen or shorten the bar by a distance $2a$, and so on, as long as we keep within the limits of elasticity —that is, do not produce a *permanent set.*

To state this law more precisely: if a longitudinal stress of p. pounds per square inch cause the length of a bar L to become $L + l$, or $L - l$, it is found that $p. \frac{L}{l}$ is a large number, which is always the same for the same material, and which has been called the Modulus of Elasticity.

The Modulus of Elasticity is the hypothetical force per square inch which would lengthen a bar by its own length, or crush it down till it became of no length at all, supposing the above law of proportionality to hold to these extreme cases. We know that such a supposition is absurd, but we shall find this number very useful if we take care never to use it in any case beyond the limits of Elasticity.

It is evident from the form of the expression $p. \frac{L}{l}$ that the above definition is correct, and that at least it will be a help to the memory.

Moduli of Elasticity.

(In pounds per square inch.)

Wrought iron	29,000,000	Copper wire	17,000,000
Cast iron	17,000,000	Oak	1,450,000
Brass	8,930,000	Fir	1,330,000
Steel (hard)	29,000,000		

The relations connecting stresses and strains in general, will be found in Thomson and Tait's "Elements of Natural Philosophy."

Example.—To find what pressure would shorten or lengthen a bar of 12,900 inches long by the distance $1\frac{1}{2}$ inch. Now, as 29,000,000 lb. per square inch would lengthen or shorten it by 12,900 inches, it requires $29{,}000{,}000 \times \frac{1\frac{1}{2}}{12{,}900}$ or 3372·09 lb. per square inch to lengthen or shorten the bar by $1\frac{1}{2}$ inch. We see that if *heat* had been allowed to lengthen the bar by this amount, it would have had to exert the enormous force of 1 ton per square inch.

Exercises.

(1.) By how much would a pound bar of wrought iron, whose diameter is 5 inches and whose length is 100 feet, lengthen under a tensile stress of 70 tons? —*Ans.* 0·46 feet.

(2.) What tensile stress must be employed to lengthen 200 feet of telegraph wire of wrought iron, $\frac{1}{4}$-inch diameter, by the distance of 2 inches?

By how many degrees ought its temperature to be raised for a similar increase of length?—*Ans.* 1,208 lb. $67^\circ{\cdot}4$.

(3.) A bar of brass, 3″ square, has its ends fixed between two immovable blocks when the temperature is 0°: what pressure will it exert against the blocks when its temperature becomes 30°?—*Ans.* 45,666 lb.

(4.) By how much will a steel piston-rod 4″ diameter and 8 feet long, shorten under a pressure of 7 tons? —*Ans.* 0·0000344 feet.

(5.) By how much will a steel piston-rod, 8 feet long and 5″ diameter, shorten under a pressure of 32 lb. per square inch on a surface of 295 square inches.—*Ans.* 0·020132 feet.

(6.) By how much will a column of oak, 7 feet long and 4 inches square, be compressed in supporting a weight of 2 tons?—*Ans.* 0·00145 feet.

(7.) What are the relative velocities of sound through wrought iron and copper if their specific gravities are 7·8 and 8·8 respectively, and the velocity of sound varies as $\sqrt{\frac{\text{elasticity}}{\text{density}}}$?—*Ans.* 1·39 : 1.

(8.) At what temperature will a bar of wrought iron break when gradually heated while prevented expanding, if the direct crushing stress on wrought iron is 38,000 lb. per square inch?—*Ans.* 10°·6.

18. **Expansion of Liquids.**—Since liquids and gases take the form of any containing vessel, we can only consider their cubical expansions, or expansions in volume; and it is deduced from experiment that whilst solids and liquids all differ in their co-efficients of expansion, *gases* have almost exactly the same co-efficient.

Now, the *apparent* expansion of a liquid is less than the *real* expansion, because the containing vessel itself gets larger.

The real expansion of mercury has been determined by experiments, in which the pressures of two columns of mercury, one hot and the other cold, in the two legs of a bent tube, balance each other; from the difference in height of the columns the expansion may be calculated.

19. In treatises on Heat, tables are given of the relative volumes certain liquids assume as the temperature increases. It will be sufficient for us to know that the co-efficient of expansion for mercury at ordinary temperatures is about ·00018, and that the expression

$$·9977 + ·000123\,t + ·00000\,33\,t^2$$

gives approximately the volume which a unit volume of water at 4° assumes when its temperature is increased to t°, t not being greater than 200°.

It is found that the expansion of a liquid for increase in temperature is always most variable near its freezing-point, and that the co-efficient of expansion increases with the temperature.

The temperature of maximum density for sea-water is $-3{\cdot}7^\circ$, and for other saline solutions varies from -17° to $3^\circ{\cdot}2$.

20. For problems on the expansion of liquids,

$$v\ a\ t$$

expresses the amount of expansion in volume where v is the volume, a the co-efficient of volumetric expansion, and t° the rise in temperature. It will be seen that, as before, the volume at t is equal to

$$\text{volume at } 0^\circ \times (1 + a\,t).$$

21. **Expansion of Gases.**—Before entering on the subject of expansion by heat, let us consider Boyle's law :—*The temperature being constant, the volume of a given quantity of a gas is inversely as its pressure or elastic force.*

This law has been shown to hold more and more accurately for vapours as they get further away from the temperatures at which they would liquefy, and to be almost absolutely true for permanent gases.

The same remark applies to the law of expansion under constant pressure. This law is, that *when a quantity of gas is heated, the amount of expansion is independent of the kind of gas, and is about* $\frac{1}{273}$*rd of the volume at* 0° *for each Centigrade degree.* For instance, 273 cubic feet of dry air at 0° C. will become 274 cubic feet at 1° C., 275 at 2° C., and indeed will

expand by one cubic foot for every degree of rise in temperature, becoming 373 cubic feet at 100°. In fact, for all gases (Art. 5) the co-efficient of volumetric expansion is $\frac{1}{273}$ or ·00367, and the law holds that

$$V = v\ (1 + \cdot 00367\ t)$$

where v is the volume of the gas at 0°, and V is the volume at t°.

It may be shown that when the pressure is unchanged

$$\frac{V}{V_1} = \frac{1 + a\,t}{1 + a\,t_1°}$$

where a = ·00367, and where V is the volume of a gas at t° and V_1 the volume at t_1°.

We have supposed the pressure of the gas to be the same hitherto at the two temperatures; but if P and P_1 are the respective pressures, since volumes vary inversely as pressures, the law becomes :—

$$\frac{V}{V_1} = \frac{1 + a\,t}{1 + a\,t} \times \frac{P_1}{P}$$

Example.—5 cubic feet of hydrogen at 63°, and the pressure of 24 inches of mercury, heated to 220°, with the pressure of 29 inches of mercury: what is the new volume?

If V is the new volume at 220°, the equation becomes

$$\frac{V}{5} = \frac{1 + 220 \times \cdot 00367}{1 + \ 63 \times \cdot 00367} \times \frac{24}{29}$$

= 1·215, hence V = 6·075 cubic feet, the volume of the expanded gas.

Example.—After a glass globe containing 2·3 cubic inches of mercury at 0° C. has been cooled to −27°, what space will remain unoccupied in the vessel? If raised to 27°, how much mercury makes its way

out by an orifice?—*Ans.* By the difference in the cubic contractions or expansions of globes of glass and mercury whose volumes are 2·3 cubic inches.

Exercises.

(1.) A cubic foot of gas at 27° is heated to 137°, and an invariable pressure is maintained by using a movable piston in a tube from the containing vessel: what is the new volume?—*Ans.* 1·366 cubic feet.

(2.) If in the last question the pressure becomes half what it was before (shown by using the proper weights in loading the piston); if it becomes twice as great, or if its new pressure is to its old as 4 : 3, what are the corresponding volumes?—*Ans.* 2·732, 0·683, 1·0245.

(3.) Air goes into a furnace at 16°, and reaches the chimney at 903°. The chimney contains 2,200 cubic feet of this hot air: what is the difference between the weight of this hot air and of an equal bulk of cold air? A cubic foot of air at 0°, and at the ordinary pressure, weighs ·0807 lb.—*Ans.* 126·5 lb.

(4.) 500 litres of hydrogen at 60, and a pressure of 750 millimetres, being cooled to 20° under 840· mm., what is the new volume?—*Ans.* 392 litres.

(5.) 100 cubic feet of steam at 100°, and 15 lb. pressure, is heated to 160° at 17 lb. pressure: what is its volume?—*Ans.* 102·4 cubic feet.

(6.) 20 cubic feet of water at 4° C. being heated to 100°, what is its volume?—*Ans.* 20·86 cubic feet.

22. Equal volumes of hydrogen, air, and certain other incondensible gases seem to expand or contract by the same amounts for the same changes in temperature. A volume of 273 cubic feet of one of these gases at 0° will expand or contract by one cubic foot for a difference of temperature of one degree.

Suppose this law to hold for all temperatures, then at $-273°$ the volume of the gas would vanish. Certainly at the temperature of $-273°$ the volume must be exceedingly small. This $-273°$ has been called the point of extreme cold, or the absolute zero of temperature (Art. 5). All ordinary readings may be changed into readings on this absolute scale (where the old $-273°$ C. is called zero) by adding $273°$.

The use of the absolute scale will help in remembering the formulæ for expansion of gases and the rule

$$\frac{V}{V_1} = \frac{1 + at}{1 + at_1} \times \frac{P_1}{P} \text{ or } \frac{V}{V_1} = \frac{273 + t}{273 + t} \times \frac{P_1}{P}$$

becomes

$$\frac{V}{V_1} = \frac{T}{T_1}\frac{P_1}{P} \text{ or } \frac{VP}{T} = \frac{V_1 P_1}{T_1}$$

where T and T_1 are the temperatures t and t_1 on the absolute scale. Thus $\frac{VP}{T}$ is always the same for the same amount of the gas.

CHAPTER III.

CALORIMETRY.

23. We have described means of indicating temperatures, but have not yet described means of measuring actual quantities of heat. Instruments which enable us to measure quantities of heat are called calorimeters.

Any effect of heat on bodies may be used in defining the unit of heat, so long as we attend to the

principle that *equal portions of the same body when acted upon in exactly the same way will have equal quantities of heat.* Thus 2 lb. of water raised in temperature from 4° to 5° will have been given twice as much heat as 1 lb. of water raised from 4° to 5°. Let then the heat necessary to raise 1 lb. of water from 4° to 5° be our *unit of heat.*

Ice at 0° receives heat for liquefaction without rising in temperature, and a quantity of heat may be measured by the weight of ice it is sufficient to meet under these circumstances. Again, water at 100° receives heat for vaporization without rising in temperature; and a quantity of heat may be measured by the weight of water it is sufficient to convert into steam.

It is not to be assumed that the quantity of heat necessary for raising 1 lb. of water from 4° to 5° is the same as that required for raising it from 5° to 6°, or from 50° to 51°; but experiment has shown that the difference is extremely small, and that m pounds of water raised in temperature through n degrees needs almost exactly the same heat as $m\ n$ pounds of water raised from 4° to 5°.

It has been established by experiment that there is no difference in the character of the heat required to melt ice, to heat water, to vaporize water, or to heat other bodies, and that the quantities are comparable in all cases, no matter what may be the source of heat. Again, it is of no consequence how a certain quantity of water is heated, whether slowly or rapidly; through whatever changes it passes, the quantity of heat necessary to raise it from one temperature to another is the same in all cases.

24. One kind of calorimeter, constructed by Laplace and Lavoisier, gives the weight of ice melted by a body as its temperature sinks to 0°. Another method of measuring heat which has been more extensively employed, is the method of mixtures, in which heat

which escapes from one body to another in contact with it is measured by the increase in temperature produced in the second body, which is commonly water. This method will often be referred to in the succeeding pages.

25. **Capacity of Bodies for Heat.**—The quantity of heat required to raise 1 lb. of water through 1° is thirty times as great as that required to raise 1 lb. of mercury through 1°; in fact, it would raise the mercury through 30°, or would raise 30 lb. of mercury through 1°. This is expressed by saying that 1 lb. of mercury has a capacity for heat, one-thirtieth that of 1 lb. of water.

If 1 lb. of mercury at 31°, and 1 lb. of water at 0°, be placed in the same vessel, the temperature of the mercury sinks through 30° to heat the water 1°, and the mixture ultimately has a temperature of 1°.

Capacity for heat of a body is the number of units of heat required to raise the body one degree in temperature, or is the number of pounds of water that would be raised one degree in temperature by the heat which raises the body one degree.

Hence, as the capacity for heat of 1 lb. of water is one unit, the capacity for heat of 1 lb. of mercury is $\frac{1}{30}$th or ·033 of a unit of heat.

The *specific heat* of a body is its capacity for heat compared with that of an equal weight of water. Hence the specific heat of mercury is ·033, and the capacity for heat of any body is equal to its specific heat multiplied by its weight in pounds.

It has been found that the specific heat of a solid is greater at a high temperature than at a low one, that of platinum at 1,200° being ·0382.

Specific Heats.

	Mean between 0° and 100°.	Mean between 0° and 300°.
Mercury	·0330	·0350
Platinum	·0355	·0355
Silver	·0557	·0611
Zinc	·0927	·1015
Copper	·0949	·1013
Iron	·1098	·1218
Glass	·1770	·1990

Olive oil	·310	
Hydrogen	3·409	At the pressure of one atmosphere.
Air	0·238	At the pressure of one atmosphere.
Carbonic oxide	0·245	At the pressure of one atmosphere.
Steam	0·480	At the pressure of one atmosphere.

The specific heat of water at any temperature may be found from the empirical formula $1 + \left\{\frac{t-4}{1,000}\right\}^2$.

With the help of the following Exercises, the student will readily understand the principles of calorimeters.

Example.—1 lb. of mercury at 96° is thrown into 1 lb. of water at 5°: what is the temperature of the mixture?

Now let x° be this common temperature. The water has risen through $x—5^\circ$, the mercury fallen $96-x^\circ$, and the heat given out by mercury must be equal to the heat received by the water. Hence, $1 \times (96—x) \times ·033 = 1 \times (x—5)$. From this equation $x = 8^\circ·93$ the answer.

Example.—1 lb. of iron at its welding-point, 1500°, is thrown into 100 lb. of water at 0° : find the temperature of the mixture. Let $x°$ be the temperature of the mixture, and since about ·122 is the mean specific heat of iron $(1,500-x) \times ·122 = x \times 100$, from which $x = 1°·93$ the answer.

Exercises.

(1.) A ton of air at 600° at the ordinary pressure is passed through oil originally at 7° C. The air is allowed to sink to 28°. How much oil will it raise to the temperature of 28°?—*Ans.* 50174 lb.

(2.) How much iron will be raised from 18° to 30° with the heat given out by three tons of water sinking from 60° to 30°?—*Ans.* 68·3 tons.

(3.) While 1 lb. of air at 700° is passing round a super-heater, it sinks to 430°. What weight of steam will this raise from 100° C. to 140° C., at the pressure of one atmosphere? And what will be the new volume of the steam, supposing steam to have $\frac{5}{8}$th the density of air at the same temperature and pressure?—*Ans.* 3·346 lb.; 69·98 cubic feet.

(4.) Twenty grammes of carbonic oxide at 680°, and at the ordinary pressure, is passed through a kilogramme of water at 0°, and escapes at the temperature of 30° : what will be the temperature of the water?—*Ans.* 3°·185.

(5.) How many units of heat are required to raise the temperature of 1 lb. of air from 20° to 600°? What will be the volume of the heated air?—*Ans.* 138·04; 39·6 cubic feet.

(6.) What will be the relative capacities for heat of the same volumes of air, carbonic oxide, steam, and hydrogen, at the same pressures, if their densities are 14·4, 14, 9, and 1 respectively?—*Ans.* All equal.

(7.) What is the capacity for heat of a cubic foot of air, and hence (Exercise 6) of a cubic foot of any other gas at the ordinary temperature and pressure.—*Ans.* 0·018212 heat units.

26. From the answers to Exercises (6) and (7) just preceding, it is seen *that a cubic foot of any gas requires the same amount of heat to raise its temperature one degree as a cubic foot of air requires, provided we have the same pressure at all times in both cases.* This amount of heat is expressed by the decimal ·018212, when the air is at the ordinary pressure and temperature.

The specific heat of a gas has been defined in one way by us, namely, as *the amount of heat necessary to raise* 1 *lb. weight of gas* 1°, *the pressure being constant.* A different evaluation of the specific heat of a gas gives it as the heat necessary to raise the temperature of 1 lb. of gas 1°, the volume being constant. Let K_p and K_v in future represent these specific heats.

Let E_t represent the elasticity of the gas when the temperature is constant. Let E_h represent the elasticity of the gas when there is no loss of heat by conduction or radiation during changes of volume (*see* Art. 2).

Suppose a quantity of heat H given to a gas of volume V. *Let there be no increase in volume.* Let p be the increase in pressure. Let p_1 be the increase in pressure when the volume is constant and the temperature is raised 1°. Then

$$\frac{K_v}{H} = \frac{p_1}{p}$$

Suppose the pressure constant. Let H be given to the gas. Let v be the increase in volume. Let v_1 be the increase in volume when (pressure remaining constant) the temperature is raised 1°. Then evidently

$$\frac{K_p}{H} = \frac{v_1}{v}\text{; hence } \frac{K_v}{K_p} = \frac{p_1}{p} \times \frac{v}{v_1}$$

But it may be shown that E_t is equal to

$$V \times \frac{\text{increase in press. for rise of } 1^\circ\text{, vol. constant}}{\text{increase in vol. for rise of } 1^\circ\text{, press. constant}}$$

$$\text{Or, } E_t = \frac{Vp_1}{v_0}$$

This valve E_t may be conceived by supposing (1) the pressure constant during rise of temperature, (2) compression taking place at a constant temperature. The law is only approximately true for 1°; it is absolutely true for an infinitely small rise in temperature.

Similarly, $E_h = \frac{Vp}{v}$; hence

$$\frac{E_t}{E_h} = \frac{Vp_1}{v_1} \div \frac{Vp}{v} = \frac{K_v}{K_p}$$

The important result has been arrived at from considering elementary principles, *that the elasticities defined above are in the ratio of the specific heats.*

27. Results of experiments on Specific Heats at constant pressure.

(1.) The specific heat of a perfect gas does not vary with the temperature (true for air, &c., approximately true for carbonic acid and vapours).

(2.) The specific heat of a gas is independent of the density. But when the weight is constant, the volume varies inversely as the density; hence the capacity for heat of a given volume of a gas varies as the density.

(3.) The capacities for heat of equal volumes of the simple and incondensible gases at the same pressure and temperature are equal; but this equality only holds approximately for gases easily condensed, such as carbonic acid. It holds, however, for compound gases formed without condensation of their elements, as hydrochloric acid and nitric oxide.

(4.) As a general rule, the same body has a lower

specific heat when solid than when liquid, and lower when liquid than when gaseous.

The heat equivalent to the work performed by a body in expanding may be called its *latent heat of expansion*. The work done in expanding may be done against external forces or against molecular forces. It has been shown that when a body is in its most dense condition, and when it is least expansible by heat, its specific heat is least. From this, and reasons deduced from thermodynamics and chemistry, it is probable that the variable part of the specific heat of solids and liquids is *latent heat of expansion*.

Regnault measured the specific heats of air and other gases at constant pressure. From considerations due to the discrepancies of Newton's law for the velocity of sound in air $\frac{E_h}{E_t}$ is known to be constant and equal to 1·408.

Let us in future call this constant N. Then $\frac{K_p}{K_v} = N$.

If we suppose that the specific heat of air is independent of the temperature, then N is really a constant, but it is probable that for all ordinary gases the value of N varies slightly at different temperatures.

Take one cubic foot of a gas at 0° and at the ordinary pressure of 15 lb. to the square inch, or 2,160 lb. per square foot. Raise the temperature to 273° without increasing the volume. The heat necessary for this is $W \times K_v \times 273$, where W = weight of a cubic foot of the gas. Now let the gas expand against the atmosphere, the resisting pressure being 2,160 lb. per square foot. Add heat until its temperature is 273° and its volume is doubled. An important principle of *Thermodynamics* makes the heat added in the last operation equal to the work done by the gas in expanding (*see* Art. 43).

This work = 2,160 foot lb. or $\frac{2160}{J}$ heat units, where J is Joule's equivalent of work for a unit of heat.

D

$$\text{Hence } W \times K_v \times 273 + \frac{2160}{J} = W \times K_p \times 273$$

$$\text{Hence} \left(K_p - K_v \right) 273\, W = \frac{2160}{J} \quad . \quad . \quad (1)$$

$$\text{also } \frac{K_p}{K_v} = N \quad . \quad . \quad . \quad . \quad . \quad (2)$$

From these two equations, given W, J, and N, it is easy to calculate K_p and K_v. Again, suppose K_p found from experiment, given W and N, we may easily determine K_v and J. Again, supposing J and K_p found from independent experiments, we may readily calculate N.

The difference between K_p and K_v has been called the latent heat of expansion of a gas.

28. **Latent Heat.**—The work done by heat in the molecules of a body is not always measurable as a rise of temperature, for heat may enter into a body doing work among the molecules without raising the temperature. A mass of ice may absorb much heat, its temperature never rising above 0°. In fact, heat may enter into ice, doing work among its molecules, converting it into water, the melting being the only indication of the entrance of heat.

Latent heat is the heat which enters into a body without increasing its temperature, being necessary for its condition, or in producing a change in the state of aggregation of its molecules.

When we say that the latent heat of water is 79°, we use an inaccurate but useful form of expression. We mean that to melt a quantity of ice at 0° without raising it in temperature, requires as much heat as would raise the temperature of an equal weight of water 79°.

1 lb. of water at 0° and 1 lb. of water at 79°, when mixed, form 2 lb. of water at 39°·5; but 1 lb. of ice at 0° and 1 lb. of water at 79° form 2 lb. of water at

0°, the water having fallen in temperature 79° to melt the ice.

In the same way we talk of the latent heat of 1 lb. of steam as 536. If we measure the amount of heat necessary to raise 1 lb. of water from 0° to 100°, it will take 5·36 times this measured heat to convert the whole of the water into steam under atmospheric pressure.

If we condense all the steam from 1 lb. of water boiling at the ordinary pressure of the atmosphere, by passing it into a large vessel of cold water, it will be found that the steam has given up 536 units of heat on condensation, besides a certain amount of heat in passing as water from 100° to the new temperature of the water in the cistern.

Regnault found that the total quantity of heat in 1 lb. of steam—that is, the number of units of heat which it is capable of giving out if liquefied at constant temperature, and then cooled to 0°—was $= 606·5 + ·305\ T$, where T is the temperature of the steam. (For Rankine's important determinations, see H in table, Art. 32.)

In working exercises, it will be remembered that the number of units of heat received must be equal to the number of units of heat given out by the parts of a system.

Example.—How many pounds of ice at 0° will be melted and raised in temperature to 9° by 90 lb. of water at 87° falling in temperature to 9°? Let there be x lb. of ice, then the heat received by the ice is

$$\overbrace{79\,n}^{\text{latent heat}} + \overbrace{9\,n}^{\text{to raise to } 9^\circ}$$

The heat given out is 90×78, hence $90 \times 78 = 79x + 9x$, from which $= x + 79·77$ lb.

In making the rough calculations involved in the

following exercises, students ought to keep in mind the circumstances under which the experiments were made, to determine the numbers 79, 536, &c. (*see* Arts. 40 and 41).

Exercises.

(1.) How much ice will be converted into water at 4° by 6 lb. of water at 70°?—*Ans.* 4·77 lb.

(2.) When 10 lb. of water is converted into steam at atmospheric pressure, how many units of heat does it take from the source of heat and surrounding bodies? —*Ans.* 5360 units.

(3.) To condense per minute 60 cubic feet of steam at 15 lb., how much water at 0° is required? The condensed water should arrive in the hot well at 50°. Suppose that a cubic foot of steam at 100° weighs as much as a cubic inch at 0°?—*Ans.* 25·39 lb.

(4.) 700 cubic feet of steam at 100° is passed into 800 lb. of water at 0°: what will be the temperature of the mixture?—*Ans.* 19°·46.

(5.) 6 lb. of steam at 122° is passed into 1,250 lb. of water at 4° and 20 lb. of floating ice at 0°: to what height will the water be raised in temperature?—*Ans.* 5°·7.

(6.) 20 lb. of saturated steam at 144° is converted into water at 30°: how many heat units has it given out?—*Ans.* 1241.

(7.) Injection water comes into the condenser at 4°, with 500 cubic feet of steam, at 8·38 lb. pressure, every minute: how much water per minute leaves the condenser at 30°? Employ table, Art. 32.—*Ans.* 224·47 lb.

(8.) In distilling water, 20 cubic feet of steam pass through the worm in one minute; the cooler has 80 cubic feet of water maintained at 30°; water is supplied to this at 0°: what is the rate of supply?—*Ans.* 14·5 lb. per minute.

(9.) 20 lb. of steam from a boiler at the pressure of $1\frac{1}{2}$ atmospheres (Art. 32) condenses in passing into 1,735 lb. of water originally at the temperature of 16°: what is the new temperature of the water?—*Ans.* 23°·2.

(10.) 600 lb. of mercury at 130° C., and 723 lb. of olive oil at 110° C., are poured into a vessel containing 165 lb. of water at 0° C., and 20 lb. of floating ice: what will be the temperature of the mixture?—*Ans.* 59°·7.

CHAPTER IV.

PRESSURE OF STEAM.

29. **Watt's Indicator.**—It will be well to consider the meaning of indicator diagrams at this early stage of our work, because we have here a practical application of the principles of co-ordinate geometry, which will be very useful when we want to trace the connection between the volume and pressure of any fluid.

The work done by a gas in expanding is measured by the increase in volume multiplied by the mean pressure during expansion. The indicator enables us to measure this work by the area of a figure which it describes.

In Fig. 1, A is screwed into a cock on the cylinder of a steam-engine. U is a small bored cylinder, in which there is a piston moving steam-tight. When the cock B is opened, steam enters U, and presses the piston up against a spiral spring contained in T S. This spring will yield least for small pressures, and, in fact, the amount of its yielding is proportional to the pressure of steam in the cylinder. The pencil-holder O P is fastened to the spring, and moves with it, holding a

pencil against the paper covering, a cylinder P. As the pressure of steam increases and diminishes, the pencil moves up and down the paper on P. Suppose now that the paper cylinder is turned under the pencil at a rate proportional to the rate of the large piston of the steam-engine: evidently the pencil will trace a curved line, since the pressure is varying at every instant.

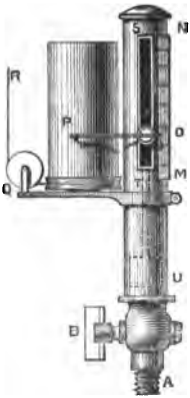


FIG. 1.

The string R enables us to turn P as the piston of the engine moves forward, whilst a concealed spring enables P to turn back again as the piston returns in its stroke.

Let the engine be supposed to have no condenser. P moves upwards, remaining near the top of the paper so long as steam enters the cylinder to keep the pressure constant behind the moving piston of the engine; when steam no longer enters from the boiler, P descends. During this motion of P, the paper moves horizontally at a regular rate, and, when unwound, shows a curved line like Fig. 2. If O P is the line traced by the pencil when the paper is motionless; and O V the line traced by the moving paper, when the pencil remains at its lowest point, and there is a perfect vacuum beneath the piston U: then the distance

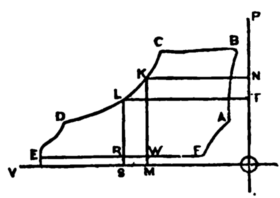


FIG. 2.

K M represents the pressure, and N R the volume of the steam in the engine-cylinder at a certain instant.

The work done by the steam in expanding to the volume L F at the pressure L S, is L R or M S multiplied by the mean pressure between K and L (Art. 2), or, in fact, the area K M S L. Hence the area O G R K L T A expresses the work done by the steam in moving the piston, and T B C O A is the work done *against* the motion of the piston in the return stroke; therefore, the area G R K L T B C represents the available work done by the piston in the complete stroke.

Indicator diagrams will be fully discussed in Book II.

30. **Vapours.**—A vapour is a gas at a temperature near to that at which condensation occurs. All bodies assume the gaseous condition at suitable temperatures. In an intensely heated furnace (that of the electric light), even carbon has been made to appear as a gas, although only in a small quantity. Most solids liquefy before becoming gaseous, but some appear to become gases at once, and are said to sublime.

This always occurs when the boiling-point of the substance at the given pressure is lower than the freezing-point for the same pressure. (*See* a paper by Professor James Thomson read before the Royal Society.)

Vapours are formed more readily *in vacuo* than in air, but for any given temperature the quantity of vapour which will form in a space from an exposed liquid is the same, whether air or other gases be present or not; the vapour being formed almost instantaneously in the second case, and requiring more or less time for formation in the first.

The pressure which this vapour eventually adds to the pressure of gases already existing in the space depends on the temperature only, and is the same, no matter what may have been the previously existing

pressure. When no more liquid will change into vapour, we may say that the space is *saturated.*

A space unsaturated with vapour may be contracted, the pressure becoming greater until the maximum pressure for that temperature is reached: after this, contraction at constant temperature causes some of the vapour to be condensed to the liquid state, as the pressure has reached its maximum, and the space has become saturated.

31. Unsaturated vapours follow approximately the laws of gases in expanding with heat. Steam, when passing along hot pipes to the engine, may be "*superheated*," and its coefficient of expansion will be found to differ very little from that of common air. By superheating steam, we increase its volume whilst its pressure is unchanged; we also render it less liable to condense in the cylinder, and we convert into steam many particles of water which are often carried over from the spray in the boiler.*

32. Saturated vapours have a different pressure for every different temperature. For instance, in a boiler, steam at the pressure of $1\frac{1}{2}$ atmospheres is formed from water at the temperature of 112°, while steam at the pressure of 1 atmosphere is formed at the temperature of 100°.

The maximum pressures of the vapour of water at different temperatures cannot be said to follow any known law, but they approximate to the following value in atmospheres :—

$$p = \left(\frac{40 + t}{140}\right)^5$$

in which the error, when t is between 80° and 225°, is very trifling.

* Priming consists in thus carrying particles of water away from the boiler. It always causes great annoyance. The use of muddy water, insufficient steam-room, carelessly constructed inside flues and pipes, &c., are the causes of priming. Superheating is one remedy for priming; for others, see Book II.

Rankine's Formulæ—

$$\log p = 6 \cdot 1007 - \frac{1517 \cdot 45}{T} - \frac{122515}{T^2}$$

when p = maximum pressure of steam in pounds per square inch.

$T = t + 273°$ the absolute temperature of the steam.

The inverse formula for calculating T when given p, is—

$$T = \frac{1}{\sqrt{\left(\frac{10 \cdot 80528 - \log p}{122515}\right)} - \cdot 006194}$$

From Regnault's experiments we obtain the following table of temperatures and corresponding maximum pressures of steam. In the third column is given the weight in pounds of one cubic foot of steam; in the fourth, the volume in cubic feet of one pound of steam; in the fifth, H_1 is the total quantity of heat (given in *foot-pounds*, see Art. 40) required to raise 1 lb. of water from 0° to $t°$, and evaporate it at $t°$. H will be found in heat units by dividing by 1390.

Temperature $t°$ C.	Pressure in lb. per square inch. p.	Weight in lb. of 1 cubic foot of steam. (Rankine.)	Volume in cubic feet of 1 lb. of steam. (Rankine.)	H_1. (Rankine.)
0	0·085	·000295	3390	842872
5	0·122	·000416	2406	844988
10	0·173	·000577	1732	847103
15	0·241	·000791	1264	849218
20	0·333	·001070	934·6	851333
25	0·452	·001431	699·0	853448
30	0·607	·001890	529·2	855563
35	0·806	·002471	404·8	857678
40	1·06	·003197	312·8	859793
45	1·38	·004099	244·0	861908
50	1·78	·005207	192·0	864024

Temperature t° C.	Pressure in lb. per square inch. p.	Weight in lb. of 1 cubic foot of steam. (Rankine.)	Volume in cubic feet of 1 lb. of steam. (Rankine.)	H_{1}. (Rankine.)
55	2·27	·006560	152·4	866139
60	2·88	·008194	122·0	868254
65	3·62	·01016	98·45	870369
70	4·51	·01250	80·02	872484
75	5·58	·01528	65·47	874600
80	6·86	·01855	53·92	876715
85	8·38	·02238	44·70	878830
90	10·16	·02685	37·26	880945
95	12·26	·03201	31·26	883060
100	14·70	·03797	26·36	885175
105	17·53	·04480	22·34	887290
110	20·80	·05260	19·03	889405
115	24·54	·06149	16·28	891520
120	28·83	·07153	14·00	893635
125	33·71	·08285	12·09	895751
130	39·25	·09559	10·48	897866
135	45·49	·1098	9·124	899981
140	52·52	·1257	7·973	902096
145	60·40	·1434	6·992	904211
150	69·21	·1630	6·153	906327
155	79·03	·1846	5·433	908442
160	89·86	·2084	4·816	910557
165	101·9	·2346	4·280	912672
170	115·1	·2635	3·814	914787
175	129·8	·2948	3·410	916902
180	145·8	·3291	3·057	919017
185	163·3	·3664	2·748	921132
190	182·4	·4068	2·476	923247
195	203·3	·4507	2·236	925362
200	225·9	·4982	2·025	927478
205	250·3	·5495	1·838	929593
210	276·9	·6048	1·672	931708
215	305·5	·6642	1·525	933823
220	336·3	·7278	1·393	935939

From the formulæ, or the table, it may be seen that the pressure increases very rapidly with high temperatures. An increase in temperature from 229° to 231° would raise the pressure in a boiler from 27 to 28 atmospheres.

It has been found that in every case the weight of a certain volume of steam at its maximum pressure is about $\frac{5}{8}$th of the weight of an equal volume of dry air at the same temperature and pressure.

The student who employs the chemical laws of gas combination will find the density given above to be absolutely correct when the vapour is very far removed from its boiling-point, and it is approximately true for ordinary steam. Rankine's more accurate determinations are given in the third and fourth columns of the table.

To find the volume of one pound of steam at its maximum pressure, without referring to the temperature, we may examine the fourth column of the annexed table; or we may employ the law just given with the formula connecting t and p to eliminate t; or we may use the empirical formula—

$$v = \frac{26{\cdot}36}{p^{\frac{16}{17}}}$$

where v is the volume in cubic feet of one pound of steam, and p the pressure in atmospheres.

Boiling.—When the pressure on the surface of a liquid is a little less than the maximum pressure corresponding to the temperature, the vapour is given off freely, and the liquid is said to boil.

Let water be gradually heated, by some means or other, in the receiver of an air-pump, with a contrivance for keeping up a determinate pressure of air in the receiver, and let arrangements be made for condensing or otherwise disposing of the vapour as it

comes off, so that the pressure on the surface of the water during the experiment may not alter. This may be effected by exposing some sulphuric acid inside the receiver. It will be found that at 100° the water boils freely under a pressure of 14·73 lb. per square inch, and explosively if the pressure is lowered. As we reduce the pressure, lower temperatures suffice for boiling the water. If by some means we could increase the pressure on the surface of the water, it would be found that a high temperature is necessary for the water before it will boil. Thus we see that *water or any other liquid boils when its vapour pressure (which alters with the temperature) is just able to overcome the pressure on its surface.*

The vapour pressures and corresponding temperatures have just been given in a table.

The ebullition of water is more retarded the greater the quantity of any saline ingredient it holds in solution; but substances mechanically suspended do not affect the boiling-point.

The boiling-points of fresh and salt water are 100° and 100°·6 respectively under ordinary atmospheric pressure. Common sea-water contains $\frac{1}{33}$ of its weight of salt. It is found that if the amount of salt be doubled (by evaporation, or otherwise) the temperature at which the water boils is raised by 0°·66; if the amount of salt is trebled, the temperature of ebullition is raised by 1°·32; and, in fact, the temperature of ebullition is raised by 0°·66 for every additional $\frac{1}{33}$ of the weight of the water of salt. Water *saturated* with common salt boils at 107°·9.

The vapour from a saline solution is found at a small distance above the liquid to have almost the same temperature as the vapour from pure water, though the solution requires a higher temperature for boiling.

Spheroidal State.—When a drop of water is thrown into a red-hot platinum cup, it is found to be separated from the metal by a cushion of steam superior in density to the air;

and as this steam conducts heat badly, the liquid gives off its vapour slowly, and at a temperature somewhat less than 100°.

33. In Art. 31 it was stated that steam followed approximately the laws of gases.

Let O V, O P in Fig. 3 be two lines at right angles to one another; let distances measured parallel to O V represent

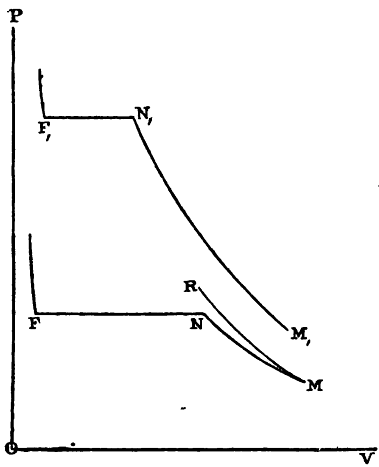


FIG. 3.

volumes, and let distances measured parallel to O P represent pressures. Let a given quantity of gas be subjected to different pressures, its temperature remaining the same. In fact, let us enclose a given quantity of gas in a cylinder whose sides are thin and conduct heat well; so that when we surround the cylinder with a large bath at a constant

temperature, the cylinder and whatever it contains will remain at a constant temperature.

An indicator diagram of a given quantity of gas will approximate to a rectangular hyperbola, for by Boyle's law the volume of a gas at constant temperature varies inversely as the pressure (or $v p =$ a constant), and points on a rectangular hyperbola fulfil this condition.

This indicator diagram may be called an *Isothermal* curve, or curve of constant temperature, for the gas in question. For a different temperature there will be a different isothermal curve, but it is evident that all curves for a perfect gas are similar.

Suppose that the gas is not perfect, and let us use steam instead of common air. Taking v cubic feet of steam at 100°, and at the pressure p, let the co-ordinates of the point M represent v and p (p being much less than the maximum pressure of steam for 100°, which is one atmosphere).

As p gets greater, the volume gets smaller, *almost* according to Boyle's law, until at length the pressure of 15 lb. is reached. The curve M N for the vapour will differ from the air curve M R. Now, the pressure of 15 lb. is the maximum pressure for steam at 100°; and attempting to apply greater pressures merely converts some vapour into water, and so the volume of one substance (which is made up of two volumes now, that of the saturated vapour and that of the liquid) gradually decreases, until at F the volume is only $\frac{1}{1680}$th of the volume at N (the volume at F is much exaggerated). Here the substance is completely liquefied, and increased pressures create so little change in volume that the rest of the isothermal curve is almost vertical.

Now, suppose the temperature had been 120°·6, and that the volume and pressure had been represented by the coordinates of M_1.

The maximum pressure for 120°·6 is two atmospheres, or about 30 lb. per square inch. The vapour at M_1 is more like a perfect gas than before, the part $M_1 N_1$ of the isothermal being much more like the hyperbola than before.*

* **Dr. Andrews' Experiments.**—*The continuity of the liquid and gaseous states of matter.*—The volumes indicated by N_1 and F_1 are nearer each other than those indicated by N and F; so that the densities

Drawing isothermals for a given weight of steam, at 100°, 101°, &c., on separate sheets of paper, and holding these sheets at equal distances asunder, parallel to each other, the similar lines O V and O P being in planes at right angles to the paper; in fact, considering that distances measured at right angles to our paper represent temperatures; a surface containing all the isothermals will determine by the three co-ordinates of any one of its points the relations between the pressure, volume, and temperature of the substance examined. A model on this principle has been constructed by Professor James Thomson for carbonic acid.

Let T, V, and P represent the temperature, volume, and pressure of a quantity of steam. Let t, v, and p represent the co-ordinates of any point on the above surface, so that t, v, and p represent the temperature, volume, and pressure of the same vapour under new circumstances. Now, the surface containing the first parts of the isothermals may be considered separately, and (Art. 21) the equation $\frac{v}{V} = \frac{1 + at}{1 + aT} \cdot \frac{P}{p}$ connects t, v, and p, since steam follows approximately the

of the vapour and of the liquid approach each other as the temperature rises.

Experiment shows that at a certain very high critical temperature the densities of a vapour and of its liquid are equal, and at this and all higher temperatures the two states of matter seem indistinguishable.

The critical temperature for carbonic acid was found to be 30°·92 C.

We will illustrate the results arrived at by Dr. Andrews by two experiments—

(1) Let a quantity of gas at 31° be gradually raised in pressure to 100 atmospheres; it will exhibit no abrupt change of state as of ordinary liquefaction. Let it be cooled to, say, 8° C.; there is still no abrupt change of state, but the carbonic acid may now be proved to be liquid; for when the pressure is gradually decreased at the lower temperature, an instant is reached at which the abrupt change from liquid to vapour becomes quite apparent.

(2.) Liquid carbonic acid at 10° C. is subjected to great pressure, and raised in temperature above 31° C.; no abrupt change has occurred. The pressure is increased until it is equal to what it was at first; there is still no abrupt change of state, but the carbonic acid may now be proved to be gaseous, for with decrease of temperature liquefaction sets in.

In fact, there is a certain temperature above which compression seems to produce no liquefaction of a vapour, and above which, reducing the pressure on the highly compressed liquid, seems to produce no vaporization, the liquid and gaseous states graduating into each other insensibly.

laws governing the expansion of a gas. This equation may be written in the form $v\,p = \frac{1 + a\,t}{1 + a\,T} \cdot P\,V$, and shows that when t is given, the isothermal is a rectangular hyperbola.

When $p = \left(\frac{40 + t}{140}\right)^5$ (Rankine's formula may be employed instead of this), the isothermal curve changes suddenly, becoming a line parallel to O V.

Maintaining the same temperature, to alter the volume and pressure of the vapour of water, *heat* must be added or taken away.

34. Now, let us examine our steam when in a cylinder fitted with a piston, *when the sides of the cylinder and the piston are quite unable to take up any heat. Hence no heat can come into or leave the steam by conduction.* Such a cylinder as this could not exist in nature, since heat can pass through all bodies, but the idea will help us in studying the nature of the connection between the volume and pressure of steam when no heat enters or leaves it.

The curve connecting v and p is now called an *Adiabatic Line.* It is to be observed, that, as before, the pressure increases as the volume diminishes. Compression produces a rise in the temperature of our steam, and as heat is prevented from leaving the body, the pressure corresponding to the new volume is greater than it would have been if the temperature had remained constant; hence an adiabatic line is "steeper" or more inclined to the line O V than an isothermal line.

From thermodynamical principles, a connection may be found between the volume and pressure (*see* Rankine's *Steam*) in every case. We shall use the approximate formula,

$$p\,v^{\frac{10}{9}} = m, \text{ or } p = m\,v^{-\frac{10}{9}}$$

m being a constant for any given quantity of heat.

CHAPTER V.

WORK AND HEAT.

35. Energy is capacity to do work, and work is done when a force overcomes resistance through any space. For instance, the force of gravity acting on a mass of one pound of anything is commonly called a force of one pound; and if the weight be allowed to move downwards any distance, whether we still hold it in our hand, or allow it to fall freely vertically, or down a curve or an inclined plane, so that there is always a distance traversed by it in a vertical direction, the *force of gravity* is said to do work. Thus, if a weight be constrained to move along the curve A O N B, or the curve A R B, or the straight line A B, or if it falls freely down the vertical line A C, we may say that gravity does work, and the amounts of work done in all these cases are the same. Again, in lifting a weight, *we* do work; for we overcome the force of gravity through a distance. Pressure in a boiler does no work on the shell, but the steam has energy; since, if properly directed, it will do work. Pressure on a piston does work when the piston yields to it.

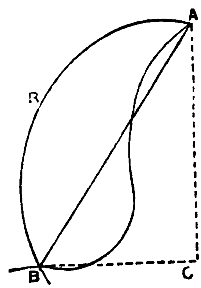


FIG. 4.

The work done by a force is measured by the product of the force into the distance through which it acts. The unit of work commonly employed is the

work done by gravity on the mass of one pound in falling through one foot, and is commonly called *one-foot-pound.*

A force of 50 lb. acting through a distance of 4 ft. is said to do 50×4, or 200 foot-lb. of work.

36. If m is the mass of a body which moves with a velocity v feet per second, the energy stored up may be expressed in foot-pounds by the formula $\frac{1}{2} m v^2$.

Given the weight of the moving body (at London), the expression $\frac{w}{32 \cdot 2}$ is the mass in a suitable form; and hence the energy stored up in the moving body is best expressed in foot-pounds by the formula—

$$\frac{w v^2}{64 \cdot 4}$$

When a force changes the velocity of a body, the work done may be calculated. A body of mass m, moving with velocity v, is acted upon by a force F in the direction of its motion, for a very short time, t. Let v_2 be the velocity at the end of time t, hence $v_2 - v_1$ is the velocity acquired. Now, force is measured by the momentum it generates in a unit of time; therefore $F = \frac{m(v_2 - v_1)}{t}$. Again, the space passed over (for v is understood to vary uniformly) is $\frac{1}{2} t (v_1 + v_2)$; hence the work done on the body, or force multiplied by space, is $\frac{1}{2} m (v_2^2 - v_1^2)$. The expression $\frac{1}{2} m v^2$ is usually termed the kinetic energy of a moving body; and we now see that the work done by a force on a moving body is equal to the increase in the kinetic energy of the body while the force is applied. This law may be extended to cases in which the force varies from time to time, and to cases in which the time has any value: indeed, it holds generally.

37. F given above may have been negative, and hence the work done in diminishing the velocity of a moving body is measured by its loss of kinetic energy

($\frac{1}{2} m v^2$). A body can do work in overcoming resistance so long as it is in motion, its capacity to do work being measured by its kinetic energy.

When a body, whose weight is w lb., is lifted through h feet in vertical height from a level floor, the work done is $w h$. Now, the body in falling under the action of gravity is able to exert a force, w, through the height h; and if we avail ourselves fully of its capacity for doing work, we shall get back from it the work $w h$. If gravity is allowed to act on the body, and the body falls freely, the work of gravity is represented by kinetic energy in the body when it reaches the floor. Thus, when a body is at rest at the height h above the floor, it has a certain capacity for doing work or for creating kinetic energy—a capacity it owes to its position; and this has been called *Potential Energy*. It loses potential energy in falling freely, by coming nearer the ground, but it gains an equivalent in kinetic energy, or else it does work; and the sum of the remaining potential energy, the gained kinetic energy, and the work done, is equal to the original potential energy.

It may be proved that, so long as no external forces act on the parts of a system of bodies, the sum of the potential and kinetic energies of all bodies in the system remains unchanged.

38. Or, to state more generally the important principle of *Conservation of Energy, the total energy of a system of bodies cannot be altered in amount by any mutual action of the bodies, although it may be changed in character.*

When, therefore, energy of any sort is given to a system of bodies, and, at the end of a certain time, the whole system is as it was at first, the energy given up to other bodies must have been equal to the energy given to the system. Now, when mechanical energy is given to a system, and

after certain changes the first state of things is restored, it is found that heat has been communicated to other bodies by the system ; hence, we infer that *heat is a form of energy.*

There are many forms of energy, as heat, electricity, and the kinetic and potential energies of moving bodies; and we find that the disappearance of one kind of energy is simultaneous with the creation of some other. In a steam-engine, heat is converted into the energy displayed in masses in motion, in the lifting of weights, in the overcoming of friction, and in the general doing of work. Again, by means of work we can produce heat, as when work is done against friction, and in percussion. The rapid compression of air produces heat sufficient to ignite gun-cotton—a common experiment in physics.

With the heat developed against friction in boring a cannon, Count Rumford was enabled to heat cold water, and eventually to make it boil; and the friction of shafts on their bearings has been known to produce sufficient heat to set fire to manufactories. I have seen a blacksmith light his fire on a morning in winter, by rapidly beating a thin steel rod on his anvil for about seven minutes, when the steel grew red-hot.

Count Rumford reasoned from his experiments on friction, and maintained that heat cannot be a material substance, as had been supposed, but must consist in motion among the particles of bodies. In fact, heat is molecular kinetic energy.

Voltaic electricity is produced by chemical action, as heat is produced by the burning of coal. A current of electricity in a wire heats the wire ; so that electricity is converted into heat. A current of electricity in an electro-magnetic machine is converted into mechanical work, such as would grind corn or move a carriage. We see, then, that wherever energy

disappears, an equivalent appears in some other shape. When a weight is stopped, after falling from a height, the visible motion plainly indicating energy is changed into an increase of motion among the molecules of the body, or gives a certain amount of heat to the body. Heat-engines enable us to reconvert part of this heat into work.

39. It was experimentally proved by Joule that 1 unit of heat is equivalent to 1390 units of work; or that the energy exerted in raising 1 lb. of water through 1° C. was equivalent to the falling of a mass of 1 lb. through a height of 1,390 feet, or the falling of 1390 lb. through a height of 1 foot.

This 1,390 is called *Joule's Equivalent*, and we always use it as a multiplier to convert heat into foot-pounds. From Joule's determinations we know that whenever work is expended, an equivalent quantity of heat, or other energy, makes its appearance.

40. It is well known that when a gas expands, it must do work against the resistance of the atmosphere, and for this reason it will be found, that to raise air to a certain temperature requires less heat when increase in volume is prevented than when the air is allowed to expand. In fact, it will be found that while the specific heat of air is ·237, when allowed to expand, it is only ·168, when its volume remains constant (*see* Arts. 26 and 27), the difference being due to a difference in the amounts of external work done in the two cases.

A gas may be changed in volume, temperature, and pressure in various ways; and although at the ends of two processes it may be in the same state, it will have required different amounts of heat for the changes, if different amounts of external work have been done in the two processes.

41. **Heat-Engines.**—At the end of Art. 38, it was stated that *part* of the heat of a hot body might be

converted into mechanical work. Heat can be converted only when being transferred from a hot body to a cold one. During the transference, some heat disappears altogether; and this which disappears is an exact equivalent of the work performed.

We are unable wholly to convert the heat which is being transferred. The greatest amount which can be converted is determined by the temperatures of the hot and cold bodies. The transference may be effected without any conversion into work, as when direct radiation and conduction occur between the hot and cold bodies.

Our natural definition of the efficiency of a heat-engine is ***the ratio of the heat converted into work to the whole heat taken from the hot body.***

Theoretical Heat-Engine—

Let the lines M N and P Q be isothermal lines for a gas of volume at the temperatures t and $t-\tau$ respectively, where τ

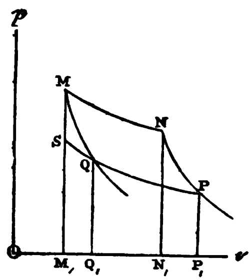


FIG. 5.

is very small. Let M represent by its ordinates the volume and pressure of the gas at first.

Let M Q and N P be adiabatic lines, very near each other,

representing indicator diagrams when the containing cylinder is non-conducting.

There are four operations to be considered in converting heat into work by means of this reversible engine, in which a perfect gas is the working body :—

(1.) Let the gas expand at a constant temperature, until at N it has absorbed heat H, and has the pressure $N N_1$.

It does work in expanding, represented by the area $M N N_1 M_1$.

(2.) Let the gas still expand, without losing or gaining heat, until its temperature is $t-\tau$, and its volume and pressure are represented by the ordinates of P.

It does work represented by the area $N P P_1 N_1$.

(3.) Let the gas be compressed at the constant temperature $t-\tau$ until its volume and pressure are represented by the ordinates of Q. In this we take away a quantity of heat = h, let us say. Work *has been done* on the gas, and is represented by the area $P Q Q_1 P_1$.

I hope it will be seen that h must be less than H, from what has already been said.

(4.) The gas is compressed without loss or gain of heat until its temperature is increased from $t-\tau$ to t, when its volume and pressure will be as they were at the beginning of the first operation. Work *has been done* on the gas, and is represented by the area $Q M M_1 Q_1$.

Altogether, then, we have given an amount of heat H to the gas at the higher temperature t, and taken an amount of heat h away at the lower temperature $t-\tau$. Besides this, we have seen that the gas performed work represented by the area $M N N_1 M_1 + N P P_1 N_1 - P Q Q_1 P_1 - Q M M_1 Q_1$ or $M N P Q$.

It may be shown that this engine is *reversible;* that is, by expending an amount of work represented by the area $M N P Q$, a quantity of heat, h, may be taken from the cold body and transferred to the hot body increased to H, where $H-h$ is the heat equivalent to $M N P Q$. Let us prove that no engine can have a greater efficiency than this reversible engine, with the same source of heat (hot body) and refrigerator (cold body).

Let A be the reversible engine. Suppose it possible that

an engine B has a greater efficiency than A. Let A take a quantity of heat H from the source, producing mechanical work w, and giving h to the refrigerator. Since A is reversible, by giving work w to the engine, heat h may be taken from the refrigerator, and heat H given up to the source.

Let B with the same heat H produce more work W, giving less heat than h to the refrigerator; and let the reversed engine A be coupled with B.

Since A receives work w, and gives H to the source of heat, and B receives H from the source, and gives out work W, the compound engine will receive *no* heat from the *source*, and yet will give out mechanical work $W-w$. But heat must have been lost to produce this work $W-w$ (Art. 38), and this heat can only have come from the *refrigerator*. Hence our compound engine performs work by means of heat drawn from the coldest body near, and according to the second principle of Thermodynamics this is absurd. Therefore, no engine can have a greater efficiency than our reversible engine.

The second law of Thermodynamics quoted above, as given by Sir W. Thomson, states that "it is impossible by means of inanimate material agency to derive mechanical work from any portion of matter by cooling it below the temperature of the coldest of surrounding objects."

All reversible engines, with the same sources and refrigerators, have the same efficiencies, whether steam or air, or other gaseous, solid, or liquid body, be used; hence the efficiency of a reversible engine is independent of the nature of the working body, depending only on the temperatures of the source and refrigerator.

When the temperatures of source and refrigerator differ by a small quantity, being t and $t-\tau$, where τ is very small, the efficiency of the perfect engine will depend on t only, when the difference τ is the same, for all perfect engines.

The efficiency divided by τ is the function discovered by Carnot; so that when τ is constant, Carnot's function depends on the temperature only.

In the above engine the efficiency is

$$\frac{\text{area MNPQ}}{JH}$$

where J is Joule's equivalent. This result may be expressed in a more general form.

The area M N P Q may be proved equal to M S $\times$ $M_1 N_1$, as the figure is almost that of a parallelogram, and M S is the difference in pressure of the gas at t and $t-\tau$, volume being constant; $M_1 N_1$ being the difference in volume which we may call v, at the temperatures t and $t-\tau$, when the pressure is constant. Let the pressure and volume of the gas at t° be P and V.

Then $\text{MS} = P - p$, where $\frac{P}{p} = \frac{1 + at}{1 + at - a\tau}$ whence $\text{MS} = P.\ \frac{a\tau}{1 + at}$ and area $\text{MNPQ} = P.\ v\left(\frac{a\tau}{1 + at}\right)$, so that the efficiency is $\frac{P a \tau v}{JH(1 + at)}$.

But J H is the work done in expanding the gas from V to V + v at the constant temperature t, and is equal to Pv, hence the efficiency is—

$$\frac{a\tau}{1 + at}$$

The efficiency divided by τ gives $\frac{a}{1 + at}$ Carnot's function, in a good form, and shows its true value when t is on the Centigrade scale.

To simplify this still further, let temperature be read on a new scale, and let $\frac{1}{a} + t$ be the new reading, corresponding to t on the old scale, then $T = \frac{1}{a} + t$, and Carnot's function becomes—

$$\frac{1}{T}$$

Now, in perfect gases, when the scale of temperature is such that $\frac{1}{a}$ is constant, and approximates to 273, $T = 273 + t$, and our new scale approximates to the absolute scale

of Art. 22. A difference in temperature τ on the old scale is the same on the new scale, and the efficiency of a reversible engine of small range τ is $\frac{\tau}{T}$.

Reversible engines of great temperature ranges may be constructed by combining a number of reversible engines of small ranges, and it may be shown easily, that if H denotes the heat taken from the source at the absolute temperature T, and H_1 denotes the heat given up again by the body at the temperature T_1, then $\frac{H}{T} = \frac{H_1}{T_1} = \frac{H - H_1}{T - T_1}$ no matter what T_1 may be.

Zero of Temperature.—J $(H - H_1)$ or $\frac{J\,H\,(T - T_1)}{T}$ is the quantity of work produced by the engine; but when $T_1 = 0$ the work is J H, and hence *all* the heat which leaves the source will be converted into work. Now, we cannot possibly have more work than this from H, hence T_1 can never become negative, and, in fact, the zero of this scale is the absolute zero of temperature.

Total Energy.—Suppose the adiabatic line M Q of Fig. 5 to be continued indefinitely until it meets the line of no pressure. A body expanding according to the conditions here represented, gives off no energy by conduction, or radiation of heat, and all its energy is converted into mechanical work; the total *intrinsic* energy being represented by the total amount of work done until complete expansion, or the area contained by the continued line M Q_1, the line M M^1, and the line of no pressure.

If the body changes its state represented in M to another, N, the amount of heat which is necessary, and the amount of work which it gives out during the change, vary with the nature of the curve M N (Art. 40); the work being represented by the area M N N_1 M_1; and the heat absorbed, by the area between M N and the two adiabatic curves. The heat H, mentioned above, is represented by the area enclosed by M N, the two adiabatics, and the line of no pressure. The heat *h*, by the area enclosed by P Q, the two adiabatics, and the line of no pressure.

Convertible Energy.—If there are a number of bodies at different temperatures and pressures in a given space, to or from which there can be no conduction of heat, and whose volume cannot change, then the greatest work which may be performed by engines, equalizing temperatures and pressures, may be calculated. This work has been called the *Entropy* of the System. When uniformity of temperature and pressure is reached in this space, no more work can be done by engines; and as radiation and conduction of heat equalize temperature without conversion of heat into work, and as communication between places of different fluid pressure will lead to equalization of pressure with the production of heat, a system of bodies is always tending to lose its entropy, and, therefore, its power of doing work.

This is the principle of *Dissipation of Energy.*

Free Expansion.—When in two non-conducting, communicating vessels, a fluid exists at different pressures, it flows from one to the other, the work expended in friction being wholly given up to the fluid, so that the intrinsic energy is unchanged. There is usually a calculable fall of temperature.*

Joule found that when air expands, the external work done is almost an exact equivalent of the heat which disappears, and, generally, that in a perfect gas no work is done against molecular forces.

The experiments of Mr. Siemens show that steam is superheated by free expansion; that is, that it is at a temperature which is above the temperature of saturation corresponding to the pressure.

It must not be forgotten, however, that there is a decided fall in temperature in this expansion of steam; and although the *intrinsic energy* of the steam is unchanged, it is now much less available than before for being converted into work in a heat-engine. In fact, it has lost *entropy*. Again, practically, the steam-passages are far from being non-conducting; hence the waste of energy which arises from throttling in steam-engines and in the exhaust of the high-pressure cylinder into the casing in most compound engines.

* See Rankine's Steam (253)

42. **Examples and Exercises.**—The work done by a given force is always measured in foot-pounds by the force in pounds multiplied by the distance through which the body is moved in feet.

The work which a moving body is able to do may be measured in London foot-pounds by the formula, $\frac{1}{2} m v^2$, where $m = \frac{\text{weight of the moving body (at London)}}{32{\cdot}2}$ and v is its velocity in feet per *second.*

Example.—What work is done in lifting 6 tons of coal through 1,542 feet vertically? Now, 6 tons = 6 × 2,240 lb.; hence the answer in foot-pounds is 6 × 2,240 × 1,542.

Example.—The mean pressure on a piston is 22 lb. per square inch; its area is 672 square inches: required the work done in 4 strokes, the length of the stroke being 5 feet. Here we have a force in pounds 22 × 672, and this acting through the distance of 20 feet in four strokes, we get for answer in foot-pounds 22 × 672 × 20.

Example.—How many heat-units are equivalent to raising 3 tons of water through 500 feet? We have here 3 × 2,240 × 500 foot-pounds, and the answer is $\frac{3 \times 2{,}240 \times 500}{1390}$; for 1,390 is Joule's equivalent (Art. 39).

Example.—A mass of 500 lb. moving with a velocity of 1,052 feet per second is gradually stopped: how many work-units will be set free?—$\frac{1}{2} \times \frac{500}{32{\cdot}2} \times 1{,}052^2$ is the answer.

Exercises.

(1.) What amount of work is involved in lifting 70 lb. 6 feet high?—*Ans.* 420 foot-pounds.

(2.) What work is involved in lifting 9,000 cubic feet

of water 46 feet? (A cubic foot of water weighs 64·4 lb.)—*Ans.* 25,833,600 foot-pounds.

(3.) What work is involved in a piston moving through 6 feet under an effective pressure of 17 lb. per square inch, its area being 1,670 square inches.—*Ans.* 170,340.

(4.) What work is involved in dragging a train weighing 700 tons along one mile of level road, the resistance to traction being constant, and $\frac{1}{320}$th of the weight?—*Ans.* 25,872,000.

(5.) How much work is performed in lifting, at a uniform rate, a train weighing 500 tons up a mile of an incline of 1 in 300, the resistance to traction on a level road being $\frac{1}{270}$th of the weight?—*Ans.* 4,161,405. (Here, if there were no friction, $\frac{1}{300}$th of the weight would be the resistance; as it is, however, $\frac{1}{270}$th + $\frac{1}{300}$th of the weight is the total resistance.)

(6.) Take the answers to the five foregoing exercises. They are expressed in work-units: express them in heat-units.

(7.) What transference of energy is involved in heating 60 lb. of mercury from its freezing-point $-39°$ to its boiling-point $-390°$?—*Ans.* 1,443·42 × 1,390 work-units.

(8.) 20 lb. of iron at the temperature of 10° falls freely into 20 lb. of water at 15° from a height of 900 feet: what will be the common temperature of the water and the mass of iron?—*Ans.* 14°·72.

43. It is to be observed that the idea of time has not yet come into our *work* questions. Now, if we can say that any agent is capable of doing a certain number of units of work in a c e r t a i n t i m e, we can compare it with any other agent in regard to rate of working. *The rates of working of two agents are in the ratio of the amounts of work done by them in the same time.*

A **Horse Power** *is defined as* 33,000 *foot-pounds in one minute.* An ordinary horse would be quite unable to do this work continuously; so that the term Horse

Power must bring before the student no other idea than that given in the definition.

The work done in one minute, divided by 33,000, gives the H.P. of any agent.

In a steam-engine the area of whose piston is A square inches, where the mean pressure of steam on the piston is P lb. per square inch, the product A P expresses the whole pressure on the piston : this, multiplied by the length of the stroke L in feet, will give the amount of work done in one stroke of the piston ; and this product, multiplied by the number of strokes in one minute, gives P L A N as the work done by the steam in one minute : hence the expression

$$\frac{\text{PLAN}}{33000}$$

gives an easily-remembered formula for the H.P. of a steam-engine.

Example.—The piston is 21" diameter, stroke 6 feet, mean pressure on the piston 16 lb. per square inch ; the engine makes 80 strokes per minute : find the H.P.

$21^2 \times \cdot 7{,}854$ = area of piston = 346·36 square inches

$$\text{H.P.} = \frac{16 \times 6 \times 346\cdot36 \times 80}{33000} = 80\cdot6144.\text{—}\textit{Ans.}$$

The number of units of work required to be done in a certain time must be equal to the number given out by the engine in the same time. Friction of the engine must at present be neglected (*see* Book II.).

Example.—What time will be taken by a steam-engine of 64 H.P. to lift 5,360 tons of water 20 feet?

Let x minutes be taken. The work which *can* be *done* by the engine in x minutes is $64 \times 33{,}000 \times x$. The work *to be done* is $5{,}360 \times 2{,}240 \times 20$; hence :—

$64 \times 33000 \times x = 5360 \times 2240 \times 20$

$$x = \frac{5360 \times 2240 \times 20}{64 \times 33000} = 114 \text{ minutes.}$$

Exercises.

(1.) A cubic mile of water is to be lifted from a depth of 2 feet in 800 hours. How many H.P. of a steam-engine is necessary? A cubic foot of water weighs 62·4 lb. nearly.—*Ans.* 11,597·6 H.P.

(2.) In a steam-engine, diam. of piston 18″, stroke 3 feet, pressure 43 lb., No. of strokes 65: find the H.P. —*Ans.* 64·6 H.P.

(3.) Piston 48″ diam., stroke 7 feet, pressure 16·3 lb., No. of strokes 23: find the H.P.—*Ans.* 114·3 H.P.

(4.) The resistance by friction, &c., to a train is a force equivalent to the weight of 600 lb. How many H.P. of the locomotive will draw the train at the rate of 35 miles an hour?—*Ans.* 56 H.P.

(5.) How many cubic feet of water will an engine of 10 H.P. raise from a depth of 150 feet in 24 hours? —*Ans.* 50,770 cubic feet.

(6.) What work is performed on a train weighing 500 tons in 3 miles of a level road, the resistance to traction being $\frac{1}{270}$th of the load? If this work were done in 6 minutes, what would be the H.P. of the engine? —*Ans.* 65,736,000 foot-pounds, 332 H.P.

(7.) Suppose the resistance to the progress of a vessel weighing 1,260 tons to be 18 lb. a ton when the speed is 6 knots, and that the resistance varies as the square of the speed: what work will be done on a vessel in 5 nautical miles, and what will be the H.P. of the engine when the speed is 12 knots? There are 6,080 feet in a nautical mile, and *a knot* is a velocity of one nautical mile per hour.—*Ans.* about 2,757,894,000; 3342·88 H.P.

(8.) What amount of work will be spent in the friction of a weight of 6 tons dragged along a level table for a length of 7 feet, when the co-efficient of friction is ·235? What will be the H.P. of an engine which would do this work in a second?—*Ans.* 22108·8; 40·19 H.P.

The co-efficient of friction is a fraction which, when multiplied on the pressure, gives the friction between two surfaces (*see* Art. 2).

(9.) If with proper arrangements a steam-engine is capable of sending out 70 cannon-balls of 6 lb. weight at the rate of 600 feet per second during every minute, what will be its H.P.?—*Ans.* 71·14 H.P.

(10.) How many heat-units per hour are involved in the idea of 62 H.P.? —*Ans.* 88,912·82.

Nominal Horse Power.—Assuming the effective pressure on the piston to be 7 lb. per square inch in all cases, and that the speed of the piston, for a given length of stroke is obtained from the following table, the nominal HP. of ordinary condensing engines may be calculated from the ordinary formula of Art. 43,

$$\frac{\text{eff. press.} \times \text{area of piston} \times \text{speed of piston}}{33000}.$$

Stroke.	Speed of piston in feet per minute.	Stroke.	Speed of piston in feet per minute.
Ft. in.		Ft. in.	
4 0	196	6 6	226
4 6	204	7 0	231
5 0	210	7 6	236
5 6	216	8 0	240
6 0	228		

The above formula may be simplified, and becomes

$$\text{Nominal H.P.} = \frac{d^2 \times \text{speed of piston}}{6000}$$

where d is the diameter of the piston in inches.

* In this table it may be seen that speed $= 120\sqrt[3]{\text{stroke}}$

It is necessary that there should be some commercial expression for the sizes and capabilities of engines, but it is evident that the nominal H.P. is one which is very defective. The nominal H.P. altogether fails to express the capability of an engine to work safely at very high speeds, and to employ high-pressure steam with good *expansion* arrangements. Again, the average pressure is now assumed to be 7·5 lb. per square inch on the Clyde, and different constants are used in other places; so that nominal H.P. has become indeterminate, and is really of very little value.

The *duty* of an engine is the amount of work done per bushel of coals burnt (a bushel of Welsh coals weighs 94 lb., and of Newcastle coals 84 lb., so that this measure of efficiency is only useful in comparing engines which burn the same kind of coal).

Given the weight w of coals burnt per indicated H.P. per hour, the duty is evidently

$$\frac{33{,}000 \times 60 \times 94 \text{ or } 84}{w}$$

an expression which may be simplified.

CHAPTER VI.

CONDUCTION AND CONVECTION.

44. **Conduction.**—When one end of a bar of copper is heated, it is found that the other parts of the bar rise in temperature, the temperatures being greater as the points are nearer the heated end. From particle to particle the heat is conducted along a bar, whether crooked or straight.

Suppose a plate, 1 square foot in surface, and 1 inch thick, to be placed in such a position that one side is maintained at a constant temperature T°, and the other at a lower temperature t° (say by means of two streams of running water whose temperatures differ). Now, if the material have nearly the same properties at T° as at t°, it will be found that the flow of heat from one side to the other is proportional to T—t.

The *conductivity* of a material is defined as *the quantity of heat which passes from a cross-section of unit area to another similar section at unit distance in a unit of time when they have unit difference of temperature.**

Fourier and Thomson have investigated the mathematical laws of conduction in bodies generally, the latter giving particular attention to the distribution of temperature within the earth. These important investigations will be found in advanced treatises on heat.

Consider the copper bar whose end is heated, after time has elapsed, and when every point has a certain temperature, which it keeps, and when, indeed, every part gives out as much heat as it receives, whether it gives out this heat by radiation or convection at the surface, or by conduction through the cross-sections.

Let the temperatures at different parts of the bar be observed.

Forbes took a similar bar of copper to that experimented upon, and, after heating the whole, measured the rapidity of its loss of heat at different temperatures by convection and radiation from the surface. His tabulated results showed the loss of heat experienced at different temperatures by unit length of the bar, and thus he determined the rate of loss at different parts of the first experimental bar, when one end was heated.

The flow of heat across any section of the bar was equal to the sum of the losses by convection, &c., on the whole

* The direction of the flow of heat is supposed to be normal to the cross-section.

length beyond this point. In this way Forbes calculated the conductivity of wrought iron, and by a very different method he determined the conductivity of certain rocks. He found that the conductivity of iron decreases as the temperature rises.

45. The following table gives the comparative conducting powers of different metals :—

Silver	100·0	Iron	11·9
Copper	73·6	Steel	11·6
Gold	53·2	Lead	8·5
Brass	23·6	Platinum	8·4
Tin	14·5		

It is a curious reminder of the marked relations between heat and electricity, that this table of relative conductivities of metal for heat is almost a counterpart of that of the relative conductivities for electricity.

In glass, wood, water, and gases, the connections among the particles are such that heat is transmitted with difficulty from one to the other. *Gases* and *liquids* are bad conductors of heat, and seem to owe most of the conducting power which they have to their property of diffusion (Art. 3). Mercury and melted metals are exceptional liquids in this respect; whereas solids are all comparatively good conductors.

The small conductivity of liquids for heat is manifested by this fact—that a red-hot ball may cause the water in a vessel to boil at the surface, whilst the temperature at a short distance below the surface remains unchanged.

We here heat the top layer of water; if we had heated the water at the bottom, the heated portions would have become lighter through expansion, and risen to the top; in the latter case, heat would have reached the surface by convection, and in the former by conduction.

Since air is a bad conductor of heat, feathers, flannel, straw, sawdust, &c., which contain small non-conducting cells of air, may be used for keeping bodies warm. Double doors having a non-conducting inter-space of air are used for furnaces for keeping in the heat.

46. If we heat a glass flask full of water, containing solid particles in suspension, by means of a gas-flame applied to the bottom, the motions of the particles will indicate currents in the water upward, as at B, in the middle, and downward, as at B, at the side (Fig. 6).

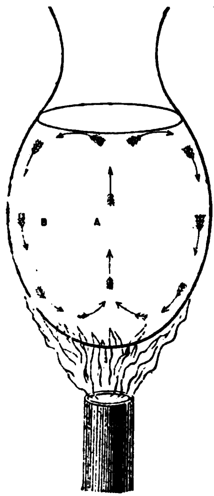


FIG. 6.

For since water at the bottom of the vessel expands on being heated, it becomes lighter, and rises to the top; in turn it gets heavy, and comes down again with other cool particles, which stream in from the sides to supply the place of the axial upward current.

Liquids and gases are mainly heated by the production of currents in this way, which has been termed *heating by convection*.

The atmosphere derives its heat from the surface of the ground, the lower layers being thus heated directly, and then ascending, forming currents or winds, in the same manner as the currents in the water of the flask are produced.

In apparatus for heating water, the arrangements must be such as to permit the vertical rise of a

heated current, and the lateral fall of a cooled current.

In heating rooms by means of water-pipes, in heating drying closets by means of hot air, in ventilating rooms, &c., it must be remembered that when fluids are heated they expand and rise, cold particles replacing those which have risen.

The heated gases of a chimney are lighter than the air outside, and hence these gases ascend with a velocity which may be calculated.

In fact, if h be the height of the chimney, t^1 the temperature of the heated gases, and t that of the air outside, calculating roughly, the velocity of ascension per second is $\sqrt{\frac{2gha(t^1-t)}{1+at}}$ where $g = 32{\cdot}2$ feet and a is the co-efficient of expansion for the gases, or $\frac{1}{273}$.

This velocity, multiplied by the cross-section in square feet, gives the volume of *heated gases* replaced every second.

The open space above the fire must not be too high nor too wide, because the cold air of the room would go up the chimney without passing through the fire.

We see, then, that when a space filled with air or water is interposed between a hot and cold body, the heat is in general conveyed, not by conduction, but by either convection or radiation.

47. **Radiation.**—As a bright body sends out rays of light in all directions, in straight lines, so does a hot body send out rays of heat in all directions in straight lines. This radiation takes place in a vacuum as well as in air, the amount of heat radiated in a given time depending on the nature and extent of the surface exposed.

The heat rays, like rays of light, consist of peculiar wave-motions, but for our purpose it will be sufficient to know two facts about them, that these rays of

radiant heat may be (1) *reflected*, and (2) *refracted* like rays of light.

Bodies are continually radiating heat. A body in the neighbourhood of others hotter than itself *receives* more heat in this way than it *gives out*, and in consequence rises in temperature.

When a body has attained a constant state with regard to heat, it is evident that it must be giving out exactly the same heat that it absorbs, both in amount and character. This consideration has led to the Theory of Exchanges,* of which the first principle has just been given, and from which may be deduced—(1), that good absorbers of radiant heat are good radiators; and (2), that good reflectors of heat are bad absorbers. This second law might have been expected, for heat which falls on a body is either reflected or absorbed, or is allowed to pass through the body without heating it (through rock salt, for instance).

Polished metallic surfaces radiate very little heat, and hence a tea-pot or polished steam-pipe cools slowly, even when in the neighbourhood of very cold bodies. Rough surfaces, in general, radiate better than smooth. Lampblack seems, of all substances, to be the best radiator of heat.

Dish-covers and the domes of a locomotive are of polished metal, to prevent *radiation;* these are cases in which hot bodies cannot readily be covered by non-conductors. Stoves in the cabins of steamboats are often, from motives of tidiness, kept bright, and hence are prevented from radiating heat. Steam-pipes are very properly kept bright when they cannot be covered.

* The Theory of Exchanges is well discussed in Balfour Stewart's Elementary Treatise on Heat."

Laws of Radiation of Heat.

(1.) The intensity of radiant heat varies inversely as the square of the distance from the heated body; when the dimensions of the radiating surface are small in comparison with the distance.

(2.) Radiant heat is more intense, as the surface on which it impinges is more nearly at right angles to the direction of the rays.

(3.) Bodies are continually giving out heat. If they possess a uniform temperature, we know that they must be *receiving* exactly the same heat that they *give out.*

(4.) The quantity of heat lost or gained in radiation or absorption by a body in a second is proportional to the difference between its temperature and that of surrounding bodies, when this difference is small.

(5.) Good absorbers are good radiators, and bad absorbers are bad radiators.

(6.) Good reflectors are always bad radiators, and bad reflectors are good radiators.

The *rate of rise in temperature* for any surface at t°, when surrounded by hot bodies at T°, will exactly agree with the *rate of cooling* when the same surface is at T°, and the surrounding bodies at t°, unless T and t differ widely.*

The rate at which a cold body rises in temperature, or the *rate of cooling* of a hot body, depends on the difference in temperature between the body in question and surrounding bodies. Again, the rate of heating of a cold body, or the rate of cooling of a hot body, will depend on the nature and extent of the exposed surface.

* We here suppose the body to have a uniform temperature, as in the case of a vessel with thin metallic sides containing water, which is kept stirred.

48. For all temperatures of the hot body less than 320°, and for all differences of temperature less than 240° between it and surrounding bodies, the following law has been proved experimentally by MM. Dulong and Petit:—The rate of cooling of a body at the temperature T, surrounded by other bodies at the temperature t, is expressed by

$$1{\cdot}0077^{t}\left(1{\cdot}007^{T-t}-1\right)$$

If we examine this, we shall find that—(1.) *The rates of cooling of a heated body are in a geometrical progression when the excess of temperature above that of the surrounding medium is constant, and the temperature of the heated body increases in an arithmetical progression.* (2.) *The common ratio is the same for the same common difference, whatever be the excess of temperature of the heated body.*

The absolute loss of heat per hour per square foot of different surfaces at the temperature T, when the enclosure is at the temperature t, has been determined and expressed by the above formula, multiplied by different constants for the different surfaces, or

$$m \times 1{\cdot}0077^{t}\left(1{\cdot}0077^{T-t}-1\right)$$

Where the values of m are *—

Polished silver	2·7
,, copper	3·31
,, tin	4·55
,, brass	5·07
Cast or wrought iron, with usual surface	67·21
Building stone, sandstone, brick	76·13
Polished limestone	78·06
Unpolished limestone	109·8
Paper of any colour	79·72
Lampblack	85·79

* From experiments made by Mr. Macfarlane, by the help of a thermo-electric junction and Thomson's reflecting galvanometer, there is reason to doubt the accuracy of the numbers given in the table. The ratio of the emissive power of polished copper to that of copper covered with soot varies from ·707 to ·690, whereas by the above table the ratio is constant and equal to ·038.

The following table gives the combined results of different experimenters. Instead of employing the formula, multiply the number corresponding to the given temperature (t), and difference ($T - t$) by 4·93 m, where m is given by the above table; and the result will be the number of units of heat lost per hour by a square foot of the given surface.

Values of $T - t$.	Values of t.				
	0° C.	20° C.	40° C.	60° C.	80° C.
240	10·69	12·40	14·35	16·72	19·48
220	8·81	10·41	11·98	13·86	16·12
200	7·40	8·58	10·01	11·64	13·45
180	6·10	7·04	8·20	9·55	11·05
160	4·89	5·67	6·61	7·68	8·95
140	3·88	4·57	5·32	6·14	7·19
120	3·02	3·56	4·15	4·84	5·64
100	2·30	2·74	3·16	3·68	4·29
80	1·74	1·99	2·30	2·73	3·18
60	1·20	1·40	1·62	1·88	2·17

To calculate the heat radiated from a square foot of polished brass, for instance, when the difference ($T - t$) between its temperature and that of surrounding bodies is 20°, and when its own temperature is, say, 220°. Here $T - t = 200$, and $t = 20$.

The number taken from the above table is 8·58, and m for brass is 5·07, hence 5·07 × 4·93 × 8·58 is the answer.

Example.—What is the quantity of heat lost per hour by a square foot of an ordinary rough surface of iron when at the temperature of 260°, surrounding bodies having the temperature of 80°?

Here $T - t = 180$, and $t = 80$, and the corresponding number in the table being 11·05, the answer is $4·93 \times 67·21 \times 11·05$ since $m = 67·21$.

49.—The loss of heat by radiation is the same in a vacuum as when bodies are separated by air or other gas, but the whole loss of heat is not the same, for a hot body may lose and a cold may gain heat by the contact and convection of surrounding gases.

The following laws have been experimentally proved by MM. Dulong and Petit:—(1.) The rate of cooling by gaseous convection is independent of the nature of the surface (this is not the case with the rate of cooling by radiation), although dependent somewhat on the form and position of the surface.

(2.) This rate of cooling varies with $(T - t)^{1·233}$ where $(T - t)$ is the excess of temperature; and it seems independent of the temperature of the heated body.

(3.) The rate of cooling depends not on the density, but on the pressure of the gas.

When two fluids, a liquid and a gas, for instance, at different temperatures, are made to pass one on either side of a separating conducting wall, such as a boiler-plate, the conduction of the boiler-plate may be considered perfect, or the plate may be supposed to be infinitely thin.

According to M. Peclet, if moving gas at $T°$ and water at $t°$ are separated by a thin plate of metal, the resistance to the passage of heat may be represented by

$$\frac{1}{A\left\{1 + B\,(T - t)\right\}} \quad . \quad . \quad . \quad (3)$$

where A and B are constants. Now, the quantity of heat (q units of heat) which will pass in an hour through one square foot of boiler-plate will be proportionate to $T - t$, and inversely proportional to the resistance ; and hence

$$q = (T - t)\, A \left\{ 1 + B\,(T - t) \right\} \dots . (2)$$

A and B have been found from experiment—

B for polished metallic surfaces	=	·0050.
B ,, rough and non-metallic surfaces	=	·0066.
A ,, polished metallic ,,	=	·9.
A ,, dull metallic ,,	=	1·58.

Rankine gives a simple formula for q, determined experimentally from the evaporation in boilers. Slightly altered, it is—

$$q = \frac{(T - t)^2}{b} \dots . (2)^*$$

where b varies from 88 to 111.

The amount of heat given to the water of an ordinary boiler, by w lb. of heated gases, is—

$$C\,W. \frac{\left(\epsilon^{\frac{A\,S}{C\,W}} - 1\right)\left\{T - t + B\,(T - t)^2\right\}}{\epsilon^{\frac{A\,S}{C\,W}} + \left(\epsilon^{\frac{A\,S}{C\,W}} - 1\right) B\,(T - t)} \dots . (3)$$

Where ϵ is the base of the Napierian or hyperbolic logarithms. A and B are given above.

W = weight of heated gases in pounds.
T = temperature of gases leaving the furnace.
t = temperature of water in boiler.
S = area in square feet of heating surface.
C = specific heat of gases. Approximately 0·22.

The second formula gives for the same heat—

$$\frac{C\,W\,S\,(T - t)^2}{S\,(T - t) + A\,C\,W} \ldots . (3)^*$$

Either of the above formulæ divided by $C\,W\,(T - t)$ will express the *efficiency* of the furnace.†

From this formula it will be found that when the area of the heating surface S has a certain value, increasing S will not materially affect the quantity of heat passing from the gases to the water. Now, as the gases in the chimney, when a chimney is employed, must be hot to produce a draught, it will be found better to allow them to escape at a high temperature than to lengthen the boiler (*see* Draught, Book II.).

50. **Combustion.**—Coal, charcoal, and timber *burn* when atoms of their carbon and other constituents unite chemically with the oxygen of the air to form oxides, such as carbonic acid and water.

Metals burn, or oxidize, less rapidly in the same manner, to form what we call rust.

It has been shown by accurate experiments, that when a combustible burns in oxygen gas or in common air, the weight of the substance formed by combustion, whether gaseous or not, is equal to the weight of the combustible, together with the weight of oxygen consumed in the combustion; in fact, whenever a combustible is burned, oxygen enters into combination, the weight of the products of combustion being equal to the sum of the weights of the separate substances before combination. This is the foundation of Lavoisier's System of Chemistry.

When carbon is burnt in air or oxygen, it disappears during

† The *Efficiency* of a furnace may be defined as the ratio—

$$\frac{T - T_1}{T - t}$$

where T_1 is the temperature of gases in the chimney.

combustion, uniting with oxygen to form the gas carbonic acid, whose weight is equal to that of the carbon and oxygen combined.

Chemical action, then, has no effect on the weights of bodies, for it has been proved that under most complicated chemical re-combinations the weights of bodies remain unchanged.

Without entering on the subject of chemical affinity, let us say that atoms of carbon have a great attraction for atoms of oxygen. When carbon burns, its atoms separate and rush towards atoms of oxygen; this rushing together of the atoms produces heat and light, as we might suppose that planets rushing together would produce heat and light.

We see, then, that before oxygen unites with any other body, some energy must be spent in separating the atoms from each other, or they will be unable to make any new arrangement; and this separation is effected by means of an increase in temperature. In fact, there is a temperature for all bodies, called the "burning-point," below which they will not burn. At this point bodies begin to burn slowly, the combustion getting more and more rapid as the temperature increases. In all cases of combustion it is necessary and sufficient to bring the combining substances into contact at a proper temperature; and in the case of gases, when these are mixed in proper proportions at a suitable temperature, the conditions for combination are perfect, and the combustion is complete.

It is known that when the temperature of carbonic oxide is too high, or about 3,000°, it will no longer combine with oxygen; and when a proper mixture of oxygen and carbonic oxide is diluted with three times its volume of any gas which does not burn with it, the temperature of 1200° will be too high for combination. Hence, when this mixture begins to burn, only one-third of it is at first involved in the combustion, the temperature of the whole rising to 3,000°. The mixture of carbonic oxide and oxygen is now diluted with carbonic acid; and as combination further dilutes it, the combustion is performed gradually at temperatures diminishing from 3,000° to 1200°. Even after this the remains of carbonic oxide will be burnt at slowly decreasing temperatures.

51. (Art. 46). In admitting air to ordinary fires, the old practice of having high open spaces above the fireplace was very defective; for, although the air mixed with the gases, the temperature of the mixture was not high enough for ignition, and the gases passed off unconsumed. According to new arrangements, the air mixes with the gases among the coals, and in the spaces surrounding the hottest parts of the fire; the proper mixture is thus obtained at a temperature high enough for combustion.

The flame from a Bunsen's burner is almost colourless, and is never used for giving out light; but from the intimate mixture of air and gas, produced before ignition, it is intensely hot. In fact, the radiation of a flame is to a great extent due to heated, unconsumed solids and dense gases; and when, as in Bunsen's burner, a proper amount of air is supplied to an ordinary coal-gas flame, these dense particles are consumed, there being complete combustion.

That it is the presence of dense gases which gives luminosity to a flame has been shown by Frankland. On increasing the pressure of the gases burnt, the luminosity is greatly increased; and candles burn with less light at the top than at the bottom of a mountain.

52. When oxygen unites with combustibles, it unites according to fixed proportions by weight. For instance, to combine with 16 lb. of oxygen 2 lb. of hydrogen are always necessary to form water. Again, to *completely* burn 12 lb. of carbon, 32 lb. of oxygen are necessary, these forming 44 lb. of carbonic acid gas.

To *partially* burn 12 lb. of carbon, 16 lb. of oxygen are necessary, these forming 28 lb. of carbonic oxide, and 28 lb. of carbonic oxide require 16 lb. of oxygen

to complete the combustion. Thus, the entrance of too little air into a furnace causes incomplete combustion.

General Chemical Composition by Weight of ordinary Coal, Coke, and Wood.

Elements.	Coal.	Coke.	Charcoal.	Wood.	
				Very dry.	Ordinary state.
Carbon .	·802	·85	·93	·51	·41
Hydrogen	·052	0	0	·05	·04
Oxygen .	·078	0	0	·42	·33
Water .					·20
Ash . .	·05	·15	·07	·02	·02

Now, 1 lb. of carbon requires $\frac{32}{12}$ lb., or 2·67 lb., of oxygen for complete combustion, and this is contained in 12·03 lb. of common air. In fact, 1 lb. of carbon requires 160 cubic feet of common air; and similarly, 1 lb. of hydrogen requires 500 cubic feet of common air for complete combustion. These volumes of air are measured at the ordinary temperature.

From these data it is easy to show from calculation that 1 lb. of coal of average composition will require in burning at least 150 cubic feet of air. As a general rule, much more air is supplied to furnaces; but, according to this, for 1 lb. of coal consumed in the furnace, 300 cubic feet of gases pass up the chimney at a temperature of about 300°.

53. The specific heats and latent heats of combustibles and of the gases formed by combustion differ so materially from one another, that in every case we must trust to special experiment for the amounts of heat given out by different bodies in burning.

The following table gives the amounts of heat from one pound of each of certain fuels when completely burnt in oxygen in a very careful manner; water, whenever formed during combustion, being supposed reduced to the liquid state :—

Substance.	Units of Heat.
Hydrogen	34000
Wood—Charcoal	8080
Coke	7000
Dry wood	4000
Ordinary wood	3000
Coal—Anthracite	8460
Coal—Dry Bituminous . .	8540 to 7655
Coal—Caking	8540
Coal—Cannel	8375
Mineral oil	12150 to 12075

The amount of heat given out by a certain weight of wood is usually the same, whatever be its character, provided it is in the same state of dryness in all cases.

Calorific Power.—When we know the composition of any fuel, we are able to calculate its total heat of combustion approximately. The hydrogen and carbon will, in general, be in considerable quantity compared with the oxygen. *Suppose all the oxygen to be already united with hydrogen, forming water: calculate the heat generated by the carbon and remaining hydrogen separately, according to the table given above, and then deduct the amount of heat necessary to convert all the water, hygroscopic or otherwise, into steam.*

The Temperature of Combustion.—If all the heat is carried off by gases from the furnace, the total heat of com-

bustion divided by W C, where W is the weight of mixed gases to which the heat is given up, and C is the specific heat of the mixture, is the difference between the temperature of combustion and that at which the gases enter the furnace. Calculations of the temperature of combustion are, therefore, easy.

From the table we find that the heating power of coal varies from 8540 to 7655.

Now, in practice there is less care taken in burning coal. (*See* Book II., Efficiency of the Boiler.)

The best experiments on the combustion of coals, were made with an ordinary single-flued Cornish boiler. The coal was thrown into the furnaces in the usual way.

Fuel.	Units of Heat by 1 lb. Coal.	Evaporative power, or the Pounds of water at 100° C. converted into steam by 1 lb. °Coal. Sometimes called *Poundage*.
Welsh Coals .	5040	9·4
Scotch Coals .	4180	7·8
Lancashire Coals	4230	7·9
Newcastle Coals.	4600	8·5

Burning combustibles give out heat in two ways :—

(1.) By radiation, depending on the difference in temperature between the combustible and the surrounding fire-box. Half the heat of burning coal is radiated.

(2.) By giving up heat to the gases which pass over and through the combustible. When the furnace of an ordinary single-flued Cornish boiler was used, the following results were obtained from the burning of 1 lb. of fuel.

Fuel.	Calorific Power.	Per cent. of heat radiated.	Per cent. heat to gases.	Cubic feet of air used at 10° C.	Cubic feet of air, &c., in chimney at 290° C.	Temperature of air leaving furnace.
Coals . .	8400	50	50	300	610	1280
Coke . .	7000	50	50	270	540	1190
Wood, ordinary state.	3000	23	77	130	280	1800

54. Voltaic electricity is produced whenever two dissimilar metals are in the same liquid touching one another. The most oxidizable metal is acted upon very rapidly; the least oxidizable metal is not acted on at all.

Pure zinc immersed in sulphuric acid is not acted upon; no bubbles of hydrogen rise to the surface of the liquid. When a piece of platinum is placed in the liquid whilst the zinc is still partially or wholly immersed, there is still no action; but when the metals are allowed to touch each other, either inside or outside the acid, bubbles of gas are instantly produced. This gas is hydrogen, set free on the surface of the platinum, and the oxygen of the water is oxidizing the zinc.

Copper, silver, iron, or tin will act like platinum. The following is a list of metals beginning with that which is the least acted upon in general by liquids :—

Gold, platinum, mercury, silver, copper, iron, tin, lead, zinc. The gold is said to be the most electro-negative, the zinc the most electro-positive.

Davy placed pieces of zinc here and there over a ship's bottom to protect the copper. The zinc oxidized

rapidly, and completely prevented oxidation of the more electro-negative metal.*

Miscellaneous Exercises.

(1.) Determine a formula connecting the temperature of superheated steam with its pressure, and its weight in lb. per cubic foot.—*Ans.* $w = \frac{273p}{T \times 14\cdot7} \times \cdot0807 \times \frac{5}{8}$ approximately.

Where p = pressure in pounds.

T = temperature on absolute scale.

The simplified formula will be found in Book II.

(2.) A mercurial barometer with a brass scale registers correctly at 0° C., the height of the barometer is 762 mm. at 16°·56: find the corresponding height under the same pressure at 0° C.—*Ans.* 760·27 mm.

(3.) Let the volume of a gas be 5 cubic feet at the pressure of 30 inches of mercury, and at 15° C.: what will be its volume at 35 inches and 72° C.?—*Ans.* 5·1355 cubic feet.

(4.) If the weight of a litre of dry air at 760 mm. and 0° C. is 1·2932 grammes, what is the weight of a litre of moist air, at 750 mm. and 15° C., if the pressure of the aqueous vapour is that which would be the maximum pressure for a temperature of 10° C.? 10° C. is called the "Dew Point."—*Ans.* 1·20401 grammes.

(5.) Suppose a body to be weighed in the air of last example and to appear to be 100 grammes. If its specific gravity is approximately 2·5, and that of the weights 9·0, find the true weight of the body.—*Ans.* 100·03492 grammes.

(6.) Find the whole amount of heat due to convection and radiation, which leaves a square foot of surface

* So many parasites gathered on the bright copper that the scheme had to be given up.

of dry chalk in one minute, the temperature being 50° C., while that of the surrounding enclosure is 14° C., the enclosure being filled with hydrogen of the pressure of 760 mm.—*Ans.* 13·958.

(7.) From Formula 3* of Art. 49, determine an expression for the efficiency of a furnace for heating water.—*Ans.* $\frac{T - T^1}{T - t} = \frac{S(T - t)}{S(T - t) + ACW}$ where T^1 is the temperature of the heated gases in the chimney.

(8.) If H is the amount of heat given to w lb. of the gases to heat them from the temperature t of the water to T of the furnace, show that the answer in (7) becomes—

$$\frac{S}{S + \frac{b C^2 W^2}{H}}$$

(9.) Show that if CW varies as FU, where F is the quantity of fuel burnt in the furnace per hour, and U is the volume of air supplied for the combustion of 1 lb. of fuel, then the answers of (7) and (8) may be written in the form—

$$\frac{BS}{S + AF}$$

Where A and B are constants. (*See* Book II., Efficiency of the Boiler.)

(10.) A pound of caking coal, containing ·88 lb. of carbon, ·052 lb. of hydrogen, ·054 lb. of oxygen, perfectly dry, is burnt in a very perfect manner: what is the total heat of combustion, not including the latent heat of the steam formed? Again, what is the evaporative power—that is, how many pounds of water at 100° C. might be converted into steam by this amount of heat?—*Ans.* 8800 units; 16 lb. of water.

(11.) As in last example, find the evaporative power

of 1 lb. of dry peat, ·58 lb. of carbon, ·06 of hydrogen, ·31 of oxygen.—*Ans.* 10 lb. of water.

(12.) Find the heat developed by the burning of 1 lb. of mineral oil, containing ·84 lb. of carbon and ·16 lb. of hydrogen.—*Ans.* 12180 units.

(13.) Find the weight of air required to burn 1 lb. of dry wood, containing ·5 lb. of carbon.—*Ans.* 6 lb.

(14.) Find accurately the weight of air required to burn 1 lb. of anthracite, containing ·915 lb. of carbon, ·035 lb. of hydrogen, ·026 of oxygen.—*Ans.* 12·13 lb.

(15.) What is the temperature of combustion of pure carbon burnt in air in a perfect manner? Heat of combustion of 1 lb. = 8080 units; weight of products of combustion = 13 lb. as may be calculated; mean specific heat known from experiment to be ·237.—*Ans.* Elevation of temperature is $\frac{8080}{13 \times \cdot 237}$, or 2540° Centigrade.

(16.) If 1·5 times the necessary air is used in Example 15, the weight of the products is 19 lb., the mean specific heat being ·237 as before. Prove that elevation of temperature = 1786 Centigrade degrees.

(17.) Suppose twice as much air is used as is necessary, mean specific heat being ·238.—*Ans.* Elevation of temperature is 1355°.

(18.) Ten litres of oxygen were measured off a 14° F. Required the volume of the gas under the same pressure at 15° C.—*Ans.* 10·95 litres.

(19.) A litre of oxygen is confined in a glass flask at 10° C., by the atmospheric pressure of 760 mm., together with that of a column of mercury 60 mm. high. Without increase of volume, what must be the new height of the mercury at 300°?—*Ans.* 1660·6 mm.

(20.) 1 lb. of ice at 0° is mixed with 1 lb. of water at 100°. What is the temperature of the mixture?—*Ans.* 10°·3 C.

(21.) 2 lb. of ice are mixed with 2 lb. of water at

79°·4. What is the temperature of the mixture?—*Ans.* 0° C.

(22.) How many pounds of steam at 100° will be required to melt 40 lb. of ice at $-4°$ C.?—*Ans.* 5·23 lb.

(23.) If the sum of the readings, for the same temperature, on a Centigrade and a Reaumur scale be 90°, what are the readings?—*Ans.* 40° R., 50° C.

(24.) The readings on C. and F. for the same temperature are in the ratio of 5 to 17: find them.—*Ans.* 20° C., 68° F.

(25.) What is the indicated H.P. of a locomotive moving at the steady speed of 30 miles per hour on a level rail, the weight of the train being 25 tons, and the resistance to traction 8 lb. per ton?—*Ans.* 16.

(26.) If a man can perform 2500 units of work per minute, in what time will he pump 100 cubic feet of water from a well whose depth is 500 feet?—*Ans.* 20·83 hours.

(27.) A body whose weight is W and temperature T, is thrown into water of temperature t and weight w. If no heat is lost by radiation or conduction, show that if t^1 is the temperature of the mixture, the specific heat of the body is

$$\frac{w\,(t^1-t)}{W\,(T-t^1)}.$$

(28.) If a gram-metre is the unit of work (the weight of one gramme lifted one metre) and the gramme-degree is the unit of heat (one gramme of water raised in temperature from 4° to 5°), show that Joule's equivalent becomes 423·55.

(29.) When the kilogram-metre and the kilogramme-degree are the two units, show that Joule's equivalent is 423·55.

(30.) If the tension of aqueous vapour in the air is f, if H is the total pressure in millimetres of mercury shown by the barometer, and t is the temperature,

show that the weight of V litres of this moist air in grammes is

$$\frac{V \times 1\cdot293}{1 + \cdot00366\,t} \cdot \frac{H - \frac{3}{8}f}{760}.$$

The weight of 1 litre of dry air at 0° C. and the pressure of 760 millimetres is 1·293 grammes.

(31.) What weight of hydrogen must be burnt to heat 109 kilogrammes of steam from 120° C. to 200° C. —*Ans.* 111·5 grammes.

(32.) One kilogramme of hydrogen is heated from 0° C. to 100° C. at the atmospheric pressure of 10330 kilos. on the square metre. How much work (in metre kilogrammes) does it do in expanding? The volume of one kilogramme of hydrogen is 11·19 cubic metres. —*Ans.* 42364·724.

(33.) What heat is required to raise one kilogramme of water from 0° C. to each of the following temperatures, and to evaporate it at each temperature: 150°, 250°, 50°?—*Ans.* 652·25, 682·75, 621·75 kilo-degrees.

(34.) What weight of saturated steam at 100° will melt one kilogramme of ice?—*Ans.* 125·6 grammes.

(35.) How much work will be required for the condensation of 100 cubic metres of saturated steam at 100° by pressure, the temperature remaining constant, neglecting the volume of the condensed steam?—*Ans.* 1,033,000 metre-kilogrammes.

(36.) What heat is necessary for raising one kilogramme of oxygen from 0° to 100°, the volume remaining constant?—*Ans.* 15·4 kilo-degrees.

(37.) Alcohol contains by weight 24 of carbon, 6 of hydrogen, and 1 of oxygen. Find the calorific power —that is, the number of units—of heat obtained by burning 1 lb. of alcohol in oxygen.—*Ans.* 6572 units, or 7200 units, assuming that the steam found was condensed.

(38.) If dried peat has the following composition by

weight :—Carbon 625·4, hydrogen 68·1, oxygen 292·4, nitrogen 14·1; find the calorific power.—*Ans.* 5521 units.

(39.) 200 cubic centimetres of hydrogen are at the pressure shown by a column of hydrogen 74 centimetres high: what is the volume when the pressure is 76 centimetres?—*Ans.* 194·7 cubic centimetres.

(40.) A closed glass vessel, which at 13° C. was well filled with air, having a pressure of 76 c. m., is heated to 559°·4: determine the pressure of the heated air.—*Ans.* 228 c. m.

(41.) A pound of mercury at 10° was mixed with a pound of water at 100°, the mixture was found to have a temperature of $97\frac{1}{8}$°: find the capacity for heat of mercury.—*Ans.* ·033.

(42.) A pound of iron at 500° is put into 10 lb. of water at 24°: what is the new temperature of the water?—*Ans.* 29°·34.

(43.) 1,000,000 cubic feet of air is lowered in temperature 10 degrees: of how much water will this increase the temperature by 3° if all the heat is communicated to the water?—*Ans.* 1026·08 cubic feet.

(44.) How many units of work must be expended in raising the materials for building a column of brickwork 100 feet high, and 14 feet square; and in how many hours will an engine of two H.P. (indicated) raise them? 1 cubic foot of brickwork weighs 112 lb. (The whole weight may be supposed to be lifted to half the height of the column.)—*Ans.* 109,760,000 units, 27·71 hours.

(45.) A body, whose weight is 10 lb., moves with a velocity of 16 feet per second; it has to overcome a constant resistance of half a pound: determine the number of feet it will describe before stopping. The kinetic energy of the body is 40 units of work. If x be the number of feet required, then $x \times \frac{1}{2}$ lb. or

$\frac{1}{2}x$ is the work done in stopping. Hence $\frac{1}{2}x = 40$, or $x = 80$ feet.

(46.) A railway truck weighing 10 tons—resistance being 8 lb. per ton—is drawn from rest by a horse; after going 300 feet it is moving at the rate of 5 feet per second: what work has been done by the horse? —*Ans.* 32750 units.

(47.) A shot weighing 6 lb. leaves the mouth of a gun with a velocity of 1000 feet per second: determine its kinetic energy, and the mean pressure exerted by the exploded powder behind it, if the length of the bore is 5 feet.—*Ans.* 93750 units of work, 18,750 lb.

(48.) Find the calorific intensity of a gas (marsh gas) burnt in oxygen, and containing 12 parts by weight of carbon for every 32 of oxygen, the calorific power being 13063. Specific heat of steam 0·4805, of carbonic acid 0·2164.—*Ans.* 7793°.

(49.) Find the calorific intensity of olefiant gas burnt in oxygen. It contains 24 parts by weight of carbon for 4 of hydrogen, and its calorific power is 11858. —*Ans.* 9136.

(50.) Find what the answers to the last two exercises would be if the combustion were performed in air. Specific heat of nitrogen 0·244.—*Ans.* 2662 and 2916. It must be remembered that in air there are 77 parts by weight of nitrogen for every 23 parts of oxygen.

(51.) 20 kilogrammes of water are heated from 4° to 5° by the combustion of hydrogen: what is the weight of the hydrogen?—*Ans.* 0·58 grammes.

(52.) What is the volume in litres of one gramme of steam at 760 mm., superheated to 200° C.? One litre of dry air at 0° and 760 mm. weighs 1·293 grammes. —*Ans.* 2·24 litres.

(53.) A glass globe of 10 litres capacity is filled with dry air at 100° C. and 760 mm.; as much water is forced in as will evaporate: what will be the tension of the moist air?—*Ans.* 1520 mm.

(54.) A mixture is required containing one volume of steam to two of air, and it is proposed to pass air through water kept uniformly at a temperature such that its vapour has the tension necessary. Find this temperature.—*Ans.* 82° C.

(55.) What is the weight of dry air contained in a glass globe of 640 cubic centimetres capacity at 546°·4 C., and under pressure of 712·5 millimetres?—*Ans.* 0·2583 grammes.

(56.) A train moving at 15 miles an hour comes to the foot of an incline of 1 in 300: if the resistance by friction is 8 lb. per ton, how far will the engine go before stopping, if the steam is shut off?—*Ans.* 1095 ft. For the kinetic energy is $\frac{W \times 22^2}{2g}$, and if x feet is the required distance then $W x\left(\frac{1}{300}+\frac{8}{2240}\right)$ is the work done against gravity and friction; hence, $\frac{W \times 22^2}{2g} = W x\left(\frac{1}{300}+\frac{8}{2240}\right)$ from which x may be calculated.

(57.) A mass weighing 1000 lb., moving at the rate of 70 feet per second, is gradually stopped by compressing steam from the volume V and pressure P: what will be the terminal volume and pressure if we suppose steam to follow the law volume × pressure = a constant? What if the law is $p \propto v^{-\frac{10}{9}}$? (That is, when the cylinder is a non-conductor of heat.)

BOOK II.

STEAM-ENGINES AND BOILERS.

BOOK II.

STEAM-ENGINES AND BOILERS

CHAPTER I.

EARLY HISTORY OF THE STEAM-ENGINE.

55. HERO (120 B.C.) employed steam in turning a reaction-wheel, where pressure is exerted on a tube by the fluid as it issues from a lateral orifice. There is no doubt but that the Egyptian priests used the pressure of vapours in performing their mysteries.

56. In the sixteenth century, essays were written on the pressure of steam in connection with the science of Pneumatics; but it was not till the seventeenth century, the time of Solomon de Caus and the Marquis of Worcester, that machines were constructed to convert the energy of steam into mechanical work.

By allowing steam of considerable pressure to fill a closed space above the surface of water in a vessel, a jet of the height of forty feet is supposed to have been obtained at this time.

57. Captain Savery, at the end of the seventeenth century, constructed an atmospheric engine. A vacuum was first obtained by filling a vessel with steam, closing

the cock, and condensing the steam by means of cold water. On turning a second cock, this vessel was connected with a pipe which opened below the surface of water in a well; and, under atmospheric pressure, the vessel might be filled with water when its height was less than 33 feet from the surface in the well. This water was afterwards raised much higher by means of steam from a strong boiler, according to the principle enunciated by the Marquis of Worcester.

Steam in contact with cold water condenses rapidly, and hence there is considerable waste of heat in employing the principle of the Marquis of Worcester; so that Savery's method of raising water to a greater height than 33 feet gave way to that of Newcomen, who used atmospheric pressure for all heights.

58. **Newcomen's Engine** (Fig. 7).—From a boiler, steam at a pressure somewhat greater than 14·73 lb., the pressure of the atmosphere, is admitted by the cock A to the cylinder B, in which the piston E moves steam-tight.

The piston had a packing of hemp round its circumference, and it was made air and steam-tight by being kept covered with water.

The piston is attached by a chain to the cross-head G of the beam G H, which moves about a central pivot. At H, a chain supports a counterbalance weight, with a long pump-rod and a bucket for lifting water.

Let the cock A be opened, C being closed.

As the cylinder is cold, much condensation occurs before the space B is filled with steam at the pressure of the boiler. When this has taken place, there is a noisy escape of steam and air by a valve called the *snifting-valve*, which opens outwards at the extremity of the pipe D. The noise produced by the condensation of escaped steam gives warning that A is to be closed, and C opened,

when cold injection-water rushes into the cylinder, condensing the steam and forming a vacuum. C is now closed; and as the snifting-valve only opens outwards, we may suppose that there is a perfect vacuum beneath the piston, which is therefore forced down with a pressure in pounds equal to its area in square inches multiplied by 14·73. This force is communi-

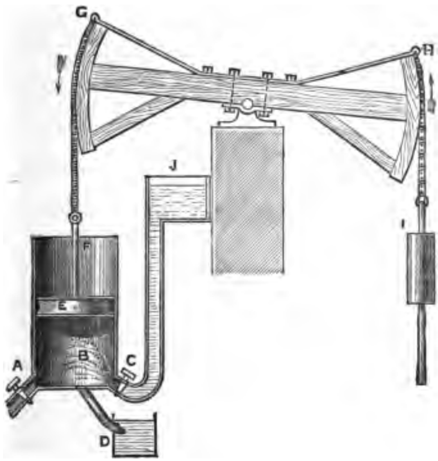


FIG. 7.

cated by the piston through the rod and chain to the beam at G, so that H is lifted with the counterbalance, weight I, the pump-rod and bucket, and a quantity of water.

When E reaches the bottom of the cylinder, A is again opened, and, with much condensation of steam,

the cylinder is heated and cleared of water, which escapes by the snifting-valve. With the help of I, the steam-pressure on the piston is now able to overcome that of the atmosphere; so that E at length reaches the top of the cylinder, and becomes filled with steam, as is duly notified by the snifting-valve. A is now closed, and C is opened; condensation occurs; a vacuum is produced; and E again descends.

A boy named Humphrey Potter designed a self-acting mechanism for closing and opening the cocks A and C. After a time this was always effected by means of projections on a vertical rod which moved with the beam, conical valves (Art. 76) being substituted for the ordinary cocks. The engines of this kind, made by Smeaton, had many improvements in the details, but they were the same in principle as that designed by Newcomen.

59. **Watt's Single-acting Engine.**—When mending a small model of Newcomen's engine for the University of Glasgow, in trying to remedy some defects in working, due to the size of the model, Watt at length came to think of a separate vessel for condensing the steam of the cylinder.

It is a principle in Physics, that when two communicating vessels containing liquids and their vapours are at different temperatures, the pressure will be the same in both, and will be that due to the lower temperature.* Hence, when a cold vessel is placed in communication with a hot cylinder filled with steam, the pressure of the vapour will almost immediately become as if the cylinder were quite cold, the steam rushing off to the colder vessel, and condensing there.

To prevent the condenser and cylinder from filling with water, Watt employed what he called an air-pump. Again, as the cylinder was to be kept hot at all times,

* This is not quite true when the condenser is small.

he covered it over at the top; and, instead of using atmospheric pressure in the descent, he employed that of steam.

Using Newcomen's beam, counterweight, and pump, Watt completely altered the character of the cylinder

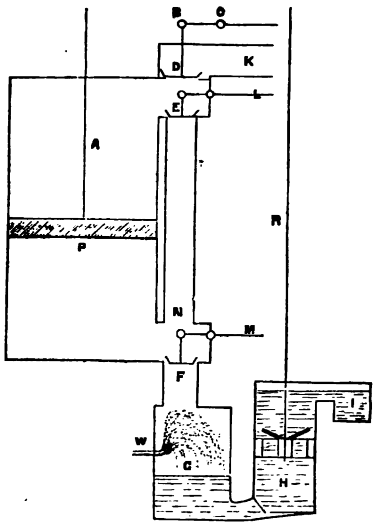


FIG. 8.

connections (*see* Fig. 8). A is the *cylinder;* P a *piston,* moving up and down, steam-tight; D, E, and F are *valves* opening upwards, provided with handles, by means of which they may be opened or closed; K is a *steam-pipe;* E N is a passage connecting the upper and

lower parts of the cylinder when E is opened; G the separate vessel for *condensing* steam; W the injection-pipe admitting cold water; H the *air-pump* (attached by a chain to the beam) separated from the condenser by the *foot-valve;* I the *hot well* from which the heated water of condensation is removed. Suppose D and F to be open, and E closed. Whatever steam existed below P has passed into the condenser; so that the pressure below the piston may be neglected. Steam now passes from the boiler by K, and the open valve D, filling A, and causing P to descend with a force proportional to the pressure in the boiler. When P reaches the bottom of the cylinder, F and D are closed, and E is opened. Thus the pressure is the same on both sides of the piston, which now rises to the top of the cylinder under the influence of the counterbalance weight. For a new stroke, F and D are again opened, and E closed; and the series of operations just described is renewed.

Injection-water enters G in spray during the stroke, and would very soon fill the condenser, but that it is drawn through the foot-valve by means of the *air-pump.* The condenser is always surrounded by cold water.* The injection-valve is opened and closed for every stroke.

Watt used modifications of the improved arrangements of Smeaton and others for distributing steam to the cylinder; a moving vertical rod, called the *plug-tree* (or the air-pump rod afterwards), carrying projections working the valves. In an engine of 1803, there were small horizontal shafts placed above each other on strong upright posts, each shaft having a handle and a short lever, which was connected to one of the three valves. There were three projections on the air-pump rod, one for depressing each handle. The valves were kept closed by means of a peculiar arrange-

* This is seldom the case in modern engines.

ment of catches, until certain weights were allowed to fall, moving the levers, and opening the valves at proper times.

The steam-jackets surrounding the cylinder, to prevent condensation of steam, were usually put together in sections jointed below, and an expansion-joint above.

Catch-pins were placed at each end of the beam to limit the motion of the piston. *Parallel motions* (*see* Art. 103) were used in the later engines of Watt, for keeping the piston-rod and great pump-rod vertical.

At the top of the pump, a large air-vessel was usually placed, so that a continuous stream of water might be produced. A small stream of air was allowed to enter the water in the pump at every stroke, to diminish the shocks of falling valves. This was also useful in keeping the air-vessel well supplied.

60. The loss of heat in Newcomen's engine was very great, through the necessity for heating and cooling the cylinder at every stroke. Watt remedied this, and still further increased the efficiency of the engine, by boring the cylinder very accurately, and using care in packing the piston, so that there was sufficient steam-tightness without the use of water.

In Newcomen's engine, when too much work was being done, the injection-cock was only partially opened, or was closed sooner than usual. Instead of this, Watt adopted the expedient of stopping the supply of steam, either by partially closing the cock D during the stroke (wire-drawing), or by completely closing it before the end of the stroke. Thus was introduced the use of expansion of steam in the cylinder described by Watt in his patent of 1782.

61. **Expansion.**—When the valve for admission is well opened during the whole descent of the piston, a cylinder, full of steam, with a pressure nearly equal to that of the boiler, flows into the condenser at every

stroke. When the valve closes at half-stroke, a certain amount of work has been done which is half the former quantity; and the steam, in expanding during the rest of the descent of the piston, continues to do work, as its pressure is always greater than that in the condenser and space below the piston. Now, this work of the second part of the stroke is considerable, and has been obtained by the use of expansion. Let us suppose that when steam expands in the cylinder to twice its volume, its pressure is exactly half what it was before, and that in general the pressure varies inversely as the volume (Art. 19).

This is not quite true, for the *adiabatic* (Art. 34) curve for steam, instead of being $p \propto v^{-1}$, as would appear from the above supposition, really approximates to

$$p \propto v^{-\frac{10}{9}}.$$

And the expansion curve is adiabatic when the cylinder is covered with some non-conducting material (*see* Art. 33, *et seq.*). In adiabatic expansion, a portion of the steam becomes condensed. When the cylinder has a *steam-jacket*, this condensation is prevented, and the curve approximates to

$$p \propto v^{-\frac{17}{16}}.$$

The effect of the steam-jacket in increasing efficiency will be discussed presently.

If, in Fig. 9, A D is the diameter and A B the length of the cylinder, the pressure during the stroke when there is no expansion may, according to some scale, be represented by any of the lengths A D, G G′, or B C.

If the steam is cut off when the piston has passed from A to G, the steam, whose volume is A G G′ D, must

expand and fill the whole cylinder, its pressure getting less and less; so that if such lines as G G′, F″ F′, E″ E, J C, represent pressures at different parts of the stroke, the curve G J is part of a rectangular hyperbola. Similarly F I and E H represent by their ordinates the pressures at different parts of the stroke, when the steam is cut off at F and E respectively. In fact, A B C D, A E H C D, A F I C D,

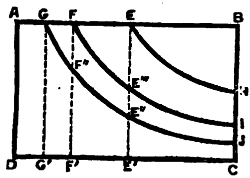


FIG 9.

&c., are imaginary indicator diagrams, and hence (Art. 29) their areas indicate the relative amounts of work performed in a stroke of the engine, when there is (1) no cut-off, (2) cut-off at half-stroke, (3) cut-off at one-fourth of the stroke, &c.

Now, the second area is nearly equal to the first; so that when expansion is allowed, a quantity of steam will perform more than three-fourths as much work as double that quantity can perform without expansion.

62. Assuming that, in the cylinder, the volume of steam varies inversely as the pressure, the work done in one stroke of the piston is

$$A q p \left(1 + \log_e \frac{l}{q}\right)^*$$

Where l = length of stroke in feet; q = A E, or A F or A G, the distance travelled over by the piston when steam is cut off; p = initial pressure per square inch; A = area of piston in square inches.

* $\log_e \frac{l}{q}$ means the Hyperbolic, or Napierian Logarithm of $\frac{l}{q}$. (A table of these logarithms is given in the Appendix.)

When there is no cut-off, the work done $= \text{A}\, p\, l$. When the cut-off is at one-fourth of the stroke, there is only one-fourth of the original quantity of steam damitted, but the work, instead of being $\frac{1}{4}\, \text{A}\, p\, l$, is thus multiplied by the quantity $1 + \log. 4$, or about 2·4; in fact, we see that one-fourth of the steam performs $\frac{3}{5}$ths of the whole work; so that by using expansion the work obtained from 1 lb. of steam is 2·4 times what it was before. This number, 2·4, has lately been called the "Indicator Co-efficient" of the engine. By cutting off at one-tenth of the stroke, it would here appear that the efficiency of the steam is increased 3·3 times—that is, that the "Indicator Co-efficient" is 3·3. But it leads to error to suppose the pressure of steam in the cylinder to vary inversely as the volume. Even with a perfect gas this would not be the case, on account of the rapid expansion.

It will be shown presently that with *jacketed cylinders* the efficiency may generally be multiplied by 2·6, when the cut-off is at one-tenth of the stroke.

As a rule, the increased expense and trouble with the large cylinders necessary, will prevent the use of so much expansion as this.

63. Expansion is valuable in another way. At the end of every stroke the piston stops, returning on its old path; and it is advisable to prepare for the sudden reversal of motion of the piston by diminishing the steam-pressure. Now, when expansion is used, the greatest pressure is exerted at the beginning of the stroke when the piston moves slowly, and when it is most advisable to get up a great velocity. The pressure diminishes gradually, until it is very little greater than that in the condenser; so that the steam experiences little difficulty in escaping by the exhaust-passages in the back-stroke. In double-acting engines, so undesirable is the maintenance of pressure, that the exhaust-port is opened before the end of the

stroke, and the exhaust-port on the other side of the piston closed, that there may be a cushion of steam for destroying the motion gradually.

The closing of the exhaust-ports may be noticed at D and F, Fig. 2.

CHAPTER II.

MODERN PUMPING AND BLOWING ENGINES.

64. **The Cornish Engine.**—This is Watt's single-acting engine of seventy years ago, improved by the use of more expansion, of higher steam-pressure, and of equilibrium double-beat valves (Art. 82). The valves are worked by projections, or *tappets*, on the *plug-tree*, different arrangements of the parts being employed by different engineers.

The steam and exhaust valves are opened by means of the *cataract*. A solid plunger falls by its own weight, or by the help of other weights, to the bottom of a small cylinder, forcing water from the space beneath it through a small orifice. As this orifice is made smaller, the plunger is made to fall more slowly. When the plunger is lifted, water is allowed to fill the space beneath it by means of a valve opening inwards. It is lifted by means of a lever, which the plug-tree pushes down; so that, when this push ceases—that is, when the plug-tree is raised rapidly from the end of the lever—the lever begins to rise, and the plunger to fall. As the lever rises, it raises a long rod with catches, and at proper instants the catches liberate weights, which move levers as they fall, and open the steam and exhaust valves. The valves are closed again by projections on the descending plug-tree.

It is evident that the size of the orifice in the cataract regulates the number of strokes per minute made by the piston; and as the orifice is usually in a small cock at the command of the engineer, it is seen that the Cornish engine is furnished with a very effective regulator.

The injection-valve is opened and closed from the cataract.

In most Cornish engines, the cylinder end of the beam is made longer than the other, for the sake of having a long stroke and great velocity of the piston with a short stroke of the pump. There are catch-pins on the beam, which strike the spring-beams of the engine-house when the travel is too great. Two parallel motions (*see* Art. 103) are usual, one for the great pump, and one for the piston-rod. It is common to find the centre of the radius bar adjustable by means of a screwed rod passing through the side of the house.

The air-pump is worked from the shorter end of the beam, so that with the condenser it is at a considerable distance from the cylinder.

Metallic packing is not yet in use for stuffing-boxes as much as it will be; for pistons its use is now universal.

A steam or hot-air jacket is always provided for the cylinders, the jacket being protected by a casing of felt, hooped with strips of wood. Even the stuffing-box is heated by means of steam, and jackets cover the top and bottom of the cylinder, as well as the sides.

A plunger-pump is always used, and there is no counterweight, the pump-rods being balanced with weighted levers. The rod is of Memel timber, and is guided in wooden frames. It has projections here and there, to enable it to get support from brackets on the sides of the shaft if it happens to break above.

65. In Browne's report, we find that in modern Cornish engines the diameter of the piston varies from 60 to 90 inches, and its stroke from 9 to 11 feet. Steam is cut off at from ·3 to ·18 of the stroke. The average pressure on the piston is 15·7 lb. per square inch. The average number of strokes per minute is 5, and the average consumption of coals is 3·4 lb. per hour for every H.P.

66. **Blowing Engines.**—These are generally constructed like pumping engines, with the substitution of a large double-acting air-compresser for the ordinary water-pump.

67. **Rotative Engines.**—Rotative engines for pumping and blowing resemble the ordinary mill-engines, to be described presently. Two engines may be coupled together, to lessen irregularities in the motion, their cranks being placed at right angles on the same shaft. A double-acting pump is usually provided for each engine. The valves are worked from cams on revolving shafts.

These engines are found to give very perfect indicator diagrams, and their efficiencies are quite equal to those of the best single-acting engines.

68. Reciprocating pumps are expensive and cumbrous; for from the use of valves the motions must be slow: they need great care in their construction and management, and seldom admit of changes in the direction of flow. Hence, it is often advisable to employ centrifugal pumps, even for considerable lifts. With great lifts, however, these pumps have little economy of power.

69. **Centrifugal Pumps and Fans.**—These depend on the principle, that when a mass of fluid is set in rapid rotation, its particles tend to fly off from the centre of rotation.

Of centrifugal pumps and fans, those designed by Professor James Thomson seem to be most efficient.

They owe their superiority to their having outside the rim of the fan a wide chamber, where the tangential velocity with which the fluid leaves the rim is gradually lost, and a power of exerting a pressure is substituted. This pressure is additional to that already given to the water by centrifugal force.

In ordinary pumps and fans this velocity is almost altogether wasted in producing eddies in the discharge-pipes.

The chief properties of this "vortex, or whirlpool, of free mobility" in the outside chamber are, "that the particles of fluid of which it is formed revolve always at velocities inversely proportional to their distances from the centre of rotation, and that each particle in moving towards or from the centre assumes of itself the velocity due to its position in the whirlpool; that, in whatever position a particle of fluid may be, the sum of its energies, corresponding to velocity, to pressure, and to height, will remain the same. Hence it follows, that as each particle gives up its velocity in passing away from the centre, it is either raised to a height corresponding to the diminution of velocity which takes place, or it becomes subject to a pressure which is capable of causing it to ascend to such a height." *

Readers may refer to Rankine's Applied Mechanics, 648—651, for formulæ connecting rate of working and velocity of delivery of fluid in pumps and fans; but it ought to be known that Dr. Rankine's treatment of the subject is objected to by some men of experience.

* Prof. James Thomson.

CHAPTER III.

VALVES GENERALLY.

72. THE entrance or escape of fluids from passages is regulated by means of cocks and valves. Valves may be moved by the fluid itself, or, like ordinary cocks, by some external agent. In pumps, the valves are opened and closed by fluid pressure; in starting an engine, the steam-valve is opened by hand; in distributing steam to the cylinder, the valves are worked by mechanism from the engine.

73. **Cocks** are usually turned by hand. A (Fig. 10) is a conical plug, having an opening of the size of the pipe in which it is placed. As the passage becomes parallel or at right angles to the direction of the pipe, the cock is opened or closed. In some water-cocks the cone is hollow, its base being beneath. Water comes up through the base, and passes into a side pipe whenever certain openings in the cone and its seat coincide with each other. Much advantage in water-tightness is here received from the hollow cone being pressed upwards against its seat.

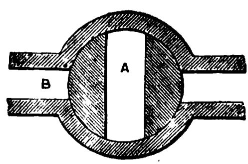


FIG. 10.

74. **The Flap Valve** has been used in common pumps from an early period. Like the door of a room opening inwards, it yields to the pressure of a fluid, and allows it to enter a chamber; but when the valve is closed by its own weight, or otherwise, and the

fluid tends to return, its pressure keeps the valve as closely shut as possible. The *foot-valve* shown at P, Fig. 25, is a flap-valve, allowing water to pass into the air-pump from the condenser when the bucket is raised, and preventing its return when the bucket is lowered. The office performed by this valve is so evident from its form, that it is often convenient to represent most of the valves in simple drawings by means of flap-valves (*see* F and I, Fig. 25).

The *face* of the valve is that surface which, by contact with its fixed seat, prevents the escape of the fluid.

It is evident that there ought to be a sufficient space about the valve in the chamber to prevent any obstruction to the passage of the fluid when the valve opens; hence these chambers, or valve-chests, are considerably wider than other parts of the piping.

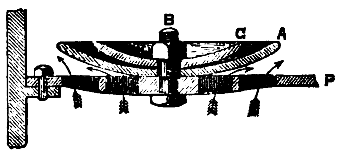


FIG. 11.

Valves and their seats ought to be of the same metal if both are metallic, on account of the galvanic action which is so apt to take place. Valves and their seats are of iron, bronze, brass, wood, india-rubber, leather, or gutta-percha; the material being suited to the nature of the fluid. When the fluid is water, for instance, hard wood is used for the valve-seats, a section across the fibres being exposed; when the fluid is steam, the valves and seats are either of brass or iron.

75. Fig. 11 shows a **Flap and Grating Valve**, used for low pressures, the fluid being water or air. A is a disc

of india-rubber, capable of covering the perforated seat P. By means of the bolt B, the guard G and the seat P are bound together at their middle parts, holding the valve A between them, not too firmly. The valve is lifted at the edges by water passing from beneath, and is pressed tightly against P when the fluid would return. The guard is often perforated. In most cases it will be thought better to use a number of separate small valves of this kind than one large one.

76. **The Conical Valve** is so called from its face and seat being parts of a conical surface. In Fig. 13 we have a peculiar combination of two conical valves, P and Q. The valve itself is flat or curved on the back, and is fastened to a spindle, or a tail, moving in guides, which insure the proper coming together of the face and seat, whilst a knob at the end of the spindle prevents too high a lift.

In the safety-valve (Fig. 51), which will presently be explained, the steam tends to pass upwards; and by its own weight, with the help of a spring or of a loaded lever, the valve is enabled to resist the steam pressure.

77. **The Divided Conical Valve.**—This was adopted to reduce the destructive effects of opening and closing large valves. A number of concentric rings, each having a valve-face on its outside, and a seat for a smaller valve on its inside circumference, gave more opening, with less elevation of each separate part, and prevented injurious shocks.

78. **The Ball Valve.**—An accurately-turned sphere is here substituted for the conical valve. In the feed-pump of Book III., B is the hollow ball resting in its seat; it is prevented from rising too high by the guard C, consisting of straps bolted to the seat. These straps must be made strong enough to resist much beating, and ought to be thick and filleted at their junction at the top.

79. **The Throttle Valve** is much used for partially stopping the flow of a fluid in a pipe. It consists of a disc of sufficient size, movable about a journal, whose bearings are placed at the extremities of a diameter of the pipe and disc. The journal is turned by means of an outside handle. When the disc is parallel to the axis of the pipe, it offers very little resistance to the passage of the fluid; but as it is turned on its axis, the opening gradually closes.

The throttle-valve is seldom used for completely stopping steam or water-passages, as the face and seat, when made accurately steam-tight, are liable to stick fast and resist opening and closing.

80. **Slide, or Sluice Valves.**—The seat and face have here considerable area, and are always in contact, sliding over each other in such a manner as to leave, at certain times, certain ports or openings in the seat uncovered by the face, so that communication is made between these openings and the valve-chest, or between this and some other port uncovered at the same time. From the necessary size of the valve, the preponderance of pressure on the back of the slide is always (unless the valve is "equilibriated") considerable, and gives rise to much friction; so that, when moved by hand, a screw must be employed, or a system of wheels moving a rack. Sometimes a small water-pressure-pump, with a cylinder and piston, is employed for the moving of very large water-valves.

The *Gridiron Valve* may be employed where a large passage for steam is to be given by means of a small motion of the valve. The seat and the slide have a number of corresponding ports at equal distances apart, so that many openings are made simultaneously.

The slide-valves commonly used for distributing steam to the cylinder of a steam-engine are described in Arts. 89—96. Gooch employs a simple steam slide-

valve in his indicator, and the use of the slide-valve is not uncommon for other purposes. A rotating slide-valve is often employed in ventilation, and the steam is admitted to the steam-pipe in some locomotives by the same contrivance. Examples of slide-valves will be found in Figs. 14, 15, 17, 21, 22.

In the *Piston Slide-Valve*, openings are covered and uncovered in the internal surface of a cylinder by

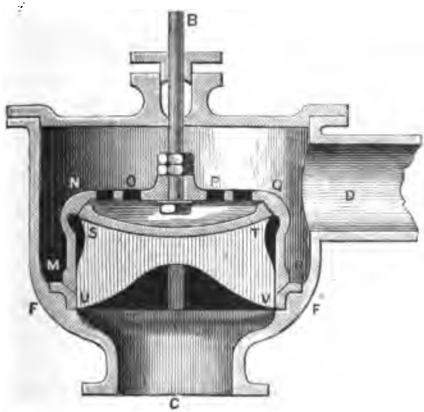


FIG. 12.

means of a piston, and communications are made as in the ordinary slide-valve.

81. In Giffard's injector (Book III.), the seat and face of the valves have considerable area, being almost complete conical surfaces.

82. **The Double-beat Valve.**—In Fig. 12, F is the valve-chest, and steam is to be admitted from D to C. M N O P Q R is the brass bell-shaped valve which has

two conical faces at M U, and immediately under N, resting on conical seats on the brass frame S T U V, which is attached by brass screws to the chest at F.

On account of the openings in the top O P, the total pressure on the valve is that due to the difference between the areas of the circles of the two seats, so that very little lifting-force is necessary. The centre line from C to B ought to be vertical, as the valve is closed by its own weight.

83. **The Equilibrium Valve** (Fig. 13).—This valve, like the last, has two conical faces at Q and P, and,

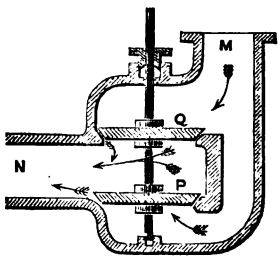


FIG. 13.

indeed, consists of two ordinary conical valves on one spindle. The steam-pressure on P tends to lift the spindle, and is balanced by the pressure on Q; so that the resistance to opening is little more than that given by the weight of the parts.

A little vessel of oil to receive the end of the spindle, and to stop it gradually, prevents the occurrence of noisy shocks. This form of valve gives great satisfaction, and is very common, especially in America.

As in the double-beat valve, a small motion gives a sufficient passage for steam.

CHAPTER IV.

DOUBLE-ACTING ENGINE.

84. IN Watt's **Single-acting Engine**, the steam acted on one side of the piston only—that is, one stroke was made during the up-and-down motion of the piston; so that half the time of this motion was devoted to the preparation for making a new stroke.

In the **Double-acting Engine**, steam acts on both sides of the piston; two strokes are made during a cycle of operations in the cylinder; the counterbalance weight is dispensed with, and the cylinders may be placed in any position.

85. Fig. 14 shows an ordinary form of the double-acting, condensing beam-engine, fitted with a locomotive slide-valve for the sake of simplicity.

It will be seen presently that, except in locomotives and marine engines, and quick-moving engines generally, the valve-motion resembles that of the single-acting engine; equilibrium valves being worked by means of tappets or projections on shafts or rods. In marine engines, which in speed and in other respects seem intermediate between locomotives and ordinary land-engines, much use is made of combined slide-valves.

E is a section of the *cylinder*, in which the *piston* C moves steam-tight. Two passages from the extremities of the cylinder alternately connect the upper and lower sides of the piston with the boiler or condenser, according to the position of the *slide-valve* in the *valve-casing* U. Whatever motion is given to the piston is communicated by the *piston-rod* D, which passes steam-tight through a *gland and stuffing-box* on the top of the cylinder, to the beam F K, by means

of the *parallel motion* J H G F I. F K is supported on strong *spring beams* N, carried by columns P, which rest on stone foundations, like the cylinder and the crank-

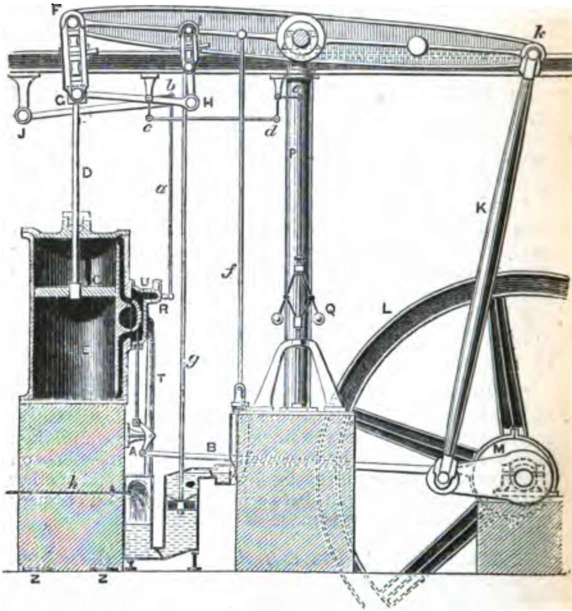


FIG. 14.

shaft pedestals. The *connecting-rod* K gives motion to the *crank* M, changing an up-and-down motion of the beam into circular motion, so that the *crank-shaft* and *fly-wheel* L revolve as the piston moves up and

down. The fly-wheel L has a heavy rim, and is difficult to stop when in motion; so that, although the pressure of the steam in the cylinder varies greatly, and would make the engine go faster at one part of the stroke than another, the velocity maintained by the fly-wheel is sensibly the same at all times, in consequence of the great mass and velocity of its rim. To change the velocity of this rim, either by an increase or diminution, a considerable amount of work must be expended, so that small perturbations of pressure in different parts of the stroke have no material effect on the motion.

Again, there are two positions of the crank and connecting-rod, called *the dead points*, at which, if an engine stops, it is necessary to turn the crank-shaft by some external means before pressure on the piston will give motion to the engine; and without the momentum of the fly-wheel the engine might stop at dead point.

There are other contrivances than the fly-wheel for regulating the motion of engines; and when two engines are coupled, in such a way that their cranks are at right angles, the irregularities of motion partly counteract each other.

A *disc* M, *keyed eccentrically* on the crank-shaft, and encircled by a ring, in which it slides, gives a backward and forward motion to the *eccentric rod* B, which moves the valves by means of the bell-crank A.

Steam reaches the *valve-casing* from the boiler by means of the pipe R, and enters the cylinder whenever the valve uncovers a *steam-port*. Methods of distributing steam will be more fully described further on. In Fig. 17 will be seen an enlarged drawing of the slide-valve. M is always in communication with the condenser, so that when the inside edges of the slide, C or A, uncover the ports, steam escapes to the condenser.

Cold water is always entering the condenser through the *injection-pipe* H, meeting the *exhaust-steam* from the cylinder, and converting it into water, absorbing heat in the process. The condensed steam and injection-water are taken through

the *foot-valve* by means of the *air-pump*, which is worked by the *air-pump rod* from a certain part of the parallel motion attached to the beam. F is the *feed-pump rod*. The *feed-pump* sends the heated water of condensation to the boiler or to a cistern.

The *governor* Q gets motion from the crank-shaft. When the velocity increases, the increased centrifugal force raises the balls, and lifts the rod P *e*, which, through the bell-cranks and rods E *d*, *c*, *b*, *a*, partially closes the *throttle-valve*, and diminishes the supply of steam.

86. **Modifications.**—Steam-engines, like that above described, in their general arrangements are quite common; but the slide-valve is too imperfect in its distribution of the steam to be employed in any but small engines, and four double-clack valves, or equilibrium valves, worked by tappets, as in some single-acting engines, take its place.

The end of the piston-rod may be furnished with a long cross-head and slides, which two rods connect directly with a crank-shaft, placed below the cylinder. This is the peculiarity of the *steeple-engine*, which is much employed in river-steamers; but the arrangement is seldom adopted in stationary engines.

Again, the cylinder may be horizontal, the end of the piston-rod having a cross-head and slides, and a single connecting-rod for the cross-head and crank. This is the common horizontal engine, in which the cylinder, piston-rod, and crank have one centre line. *Stationary, horizontal, non-condensing engines* in many ways resemble the locomotive, which will be described in Book III. *Horizontal condensing engines* differ very much in their methods for working the air-pump, and in the positions of their condensers. In some cases, one arm of a large bent lever works the pump, the other arm being slotted, and working directly on a pin on the cross-head. Again, this second arm is sometimes moved by means of a small connecting-rod from the crank-pin, which is made much longer than usual.

The piston-rod is dispensed with in *trunk-engines*.

The connecting-rod is dispensed with in *oscillating engines*.

Other modifications of the double-acting steam-engine will be described in Book IV. We find it convenient to intro-

duce at this place a notice of engines provided with cylinders for separate expansion.

Compound Engines.—The mechanical action of the steam is begun in one cylinder, and ended in one or more larger cylinders. In fact, high-pressure steam enters a cylinder whose piston is small, being cut off, and expanding in the usual manner: at the end of the stroke it enters a larger cylinder, there completing its expansion. Thus, in the small cylinder the back-pressure, or pressure behind the piston, is always the same as the forward-pressure in the large one; whereas, the back-pressure of the large one is that of the condenser. The whole amount of steam employed at each stroke is that which fills part of the small cylinder before *cut off*, and the volume of the steam, when most expanded, is that of the large cylinder, together with that of the steam-passages between.

Compound engines usually belong to one or other of two classes. (1.) Those in which the motion of the first piston is always in the opposite direction to that of the second. (2.) Those in which the motion of the first piston is sometimes in the opposite direction to that of the second, and sometimes in the same direction; there being an intermediate *casing* filled with steam.

In *McNaught's* beam-engines two cylinders are on opposite sides of the beam, and a steam-pipe connects the tops, whilst another connects the bottoms.

In *Elder's* marine-engine, the cylinders lie side by side, their pistons moving in opposite directions, working cranks on the same shaft. Hence, contiguous ends may be connected as before. Elder employs either one or two expansion cylinders.

In *Craddock's* engines, the two cylinders lie side by side, their pistons moving in opposite directions during the greater parts of their strokes, but the smaller a little in advance of the larger, to help in passing the dead points. In this case the steam is compressed again behind the small piston during a small part of the return-stroke.

In *Rowan's* engine, the high-pressure cylinder is within the larger or expansion cylinder. There are separate piston-rods, but they move the same cross-head.

In late volumes of *The Engineer* and of *Engineering*, the reader will find drawings of some good examples of compound engines.

The principal advantage arising from their use is, that there may be a greater equalization of strains than is usual in the mechanism, as it is a small piston which is subjected to the greatest steam-pressure. Again, it will usually be found sufficient to surround the small high-pressure cylinder only with a steam-jacket, so that there is less loss of heat by radiation. Lastly, by the balancing of pressures the friction at the crank-shaft bearings may be destroyed. These advantages are greatest when high grades of expansion are necessary.

In the arrangement of Randolph and Elder, where a high-pressure is placed between two low-pressure cylinders, the balancing is almost perfect.

The advantages are greatest when the work per stroke done in the *high*-pressure and *low*-pressure cylinders is the same.

Rankine, in a paper in *The Engineer*, gave the following empirical rule:—The ratio of the volumes of the low-pressure and high-pressure cylinders is $\sqrt[3]{r}$ when there is no reservoir, r being the ratio of expansion.

When there is a reservoir, the above ratio of volumes becomes $\sqrt{r}$.

In *disc-engines*, a disc, at whose centre is a ball, vibrates in a cylinder, which is bounded on the sides by a spherical zone, and at the ends by two cones; whole vertices are at the centre of the ball. There is a fixed partition in the cylinder, to which a radial slot on the disc is fitted. Two of the spaces into which the cylinder is divided get larger as the other two get smaller under steam-pressure, and the motion thus given to the central ball is transmitted to a crank by means of a long pin.

CHAPTER V.

THE CYLINDER.

87. Fig. 15 represents the section of a small cylinder, and gives a good idea of the general arrangement of parts. In larger examples, the metal is thinner in proportion to the size,

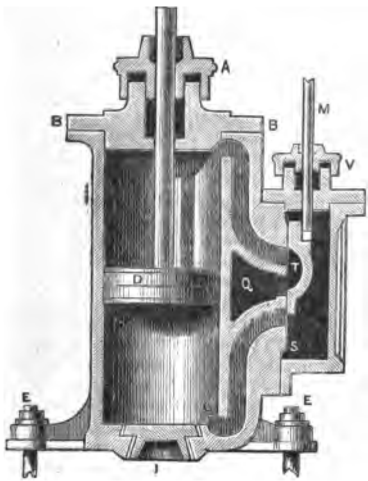


FIG. 15.

as it is necessary to give additional thickness to the metal of all cylinders for the sake of stiffness; and this addition must be greater proportionally in small than in large examples.

The metal is usually toughened cast iron, and it varies in thickness from $\frac{1}{40}$th of the diameter in low-pressure, to $\frac{1}{16}$th of the diameter in high-pressure cylinders.

The *clearance*, or distance from F and G to the ends of the cylinder, is introduced to allow for slight lengthenings or shortenings of the piston-rod, or other connections, and is practically the same for all sizes of cylinder.

The steam-passages are as wide as possible, that the back-pressure may be low; and as short as possible, for the sake of steam which would be unnecessarily wasted at every stroke. When double-clack valves are used, as in single-acting engines, or when the long D valve is employed, the ports may readily be made wide and straight and short.

Saturated steam always condenses when, in a cylinder well covered with non-conducting materials, it expands, *doing work*. By applying the principles of Art. 41 to the case of steam, this may be shown for the smallest expansions. The loss of energy from this cause would be inconsiderable, were it not that water-spray has a tendency to take heat from the entering steam, giving it out again to steam at a lower pressure and temperature, without producing much useful effect; so that unjacketed cylinders, however well protected with non-conducting coverings, cannot be so efficient as similar cylinders provided with steam or hot-air jackets. This loss of energy is not taken into account in the calculations on efficiency at the end of the book, as no scientific experiments have yet been made on the influence of the spray.

If steam were a good conductor of heat, *very little* condensation would be necessary, but usually there is much more than would be represented by the heat which would suffice to prevent condensation altogether. The steam-jacket is able to give to the steam what will just prevent condensation. Certainly, it gives nearly the same amount to the cylinder during exhaust; but in both cases the quantity of heat is itself insignificant, its importance arising from its power of preventing condensation.

Hence all parts of an engine about the cylinder ought, if possible, to be provided with steam-jackets. Steam is sometimes, when the velocity of the piston is small, allowed

to enter even into the stuffing-box around the piston-rod, that heat may be conducted to the expanding vapour.

In ordinary mill-engines the cylinder and valve-casing are covered with felt, which is a bad conductor of heat, the felt being cased with long, narrow pieces of well-dried timber, bound round with hoop-iron.

Metallic packing is thought to be better than any other for the stuffing-box, and is certainly best when the piston-rod works truly. Rings, each cut at one place, are sprung on the piston-rod, and are surrounded with packings of hemp.

Sheet-brass, backed by hemp, is found to work very well, giving a fine polished surface to the rod.

Cylinders ought to be bored in positions similar to those in which they will be placed in the engine. Cylinders bored horizontally become oval when placed upright, and *vice versâ*. Strains are often introduced in bolting cylinders to their places before boring.

When they are being bolted into their places, strains of all sorts must be carefully guarded against.

The tool ought not to travel faster than about five feet per minute, as a greater speed wears it too much, and causes too great a variation in the diameter of the cylinder.

Cylinders are often ground with fine emery on blocks of lead, cast to the proper shape; but careful, slow boring will be found in most cases to give a sufficiently smooth surface.

When, as with the long D valve, a long valve-casing is used, it ought to be provided with an expansion-joint, as it is always heated before the cylinder, and unequal expansion might give rise to injurious strains.

The *valve-seat*, when not cast in, is carefully attached by means of a metallic joint. The casing of the *blow-through valve*, whenever this is provided, is generally cast on the cylinder.

Escape-valves for condensed water are provided for the cylinder-covers. They open when the pressure on the valve is much greater than the initial pressure of the steam, and sometimes prevent the breaking of the cover by the piston towards the end of the stroke. Methods of conducting away this escaping water have sometimes to be contrived.

The most important parts of the engine are those which regulate the distribution of steam; and the most important contrivance for distributing steam is the slide-valve, whose motion we now proceed to describe.

CHAPTER VI.

THE DISTRIBUTION OF STEAM.

88. WHETHER the eccentric-rod moves the valve-rod directly, or by means of levers, the motions given up during its stroke have the same characters, being slow at the beginning, quick just after the middle, and slow again at the end. In this and succeeding articles we shall consider the effect of this peculiar motion on the distribution of steam by means of the slide-valve.

The Eccentric.—Fig. 16 represents an Eccentric. B is the crank-shaft on which the cast or wrought iron disc A is keyed, or held by pinching-screws, and the disc is surrounded by the wrought-iron strap S R V T, bolted together at F and R, which allows it to slide. A is usually lightened by being cut away in places, and is commonly attached in two halves, which are bolted together.

The straps, which are usually of wrought iron, with or without inserted brasses, clasp the outside of the disc, as shown in section at W; or they may be kept in their places laterally by means of a square fillet on the disc; or they may clasp a slight convexity of the disc by a similar concavity; or, again, they may be partially enclosed between side flanges on the disc.

One of the straps is in a piece with the eccentric-rod S M, which at M works the slide-valve rod, directly or

through a lever. M requires to be steeled, or to have a brass bush.

89. In marine-engines, when no link-motion is used, it is common to have the eccentric loose on the shaft, for the purpose of reversing the motion of the engine, and the rod has merely a notch, or *gab*, at M, for facility in throwing the eccentric out of gear.

If O is the centre of the shaft, and K that of the disc, it is evident that the motion given by the eccentric is exactly the same as that given by a small crank, of length OK; and in many cases, where the relations of the motions of the piston and valve have to be considered, we shall have to employ a small crank instead of the eccentric. This little crank will be nearly at right angles to the engine-crank, so that the valve is nearly in mid-stroke when the piston reaches the end of the cylinder. Evidently OK is half the throw, or stroke, of the valve.

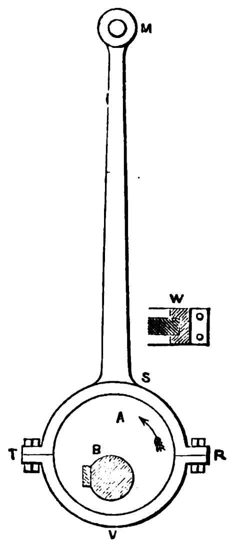


FIG. 16.

It may be shown, that when on a crank-pin a small pinion is placed, gearing into a wheel of twice its size on the crank-shaft, the motion of a point in the circumference of this wheel will give better results in regulating the motion of the valve than the use of lap and the ordinary eccentric will do. This arrangement would be equivalent to using a tappet of the shape of an epicycloid. The relative sizes of the gearing-

wheels may be modified; but further consideration of the problem must be left to the student, for whom it will be a very useful exercise.

90. **The Locomotive, or Ordinary Three-ported Valve.**—The eccentric gives a motion to the valve which is slow at the ends of the valve-stroke, and quick in the middle; this motion corresponds to the motion of the piston, as we have just seen.

It is necessary that the slide-valve should be long enough to cover both steam-ports at once, otherwise, at

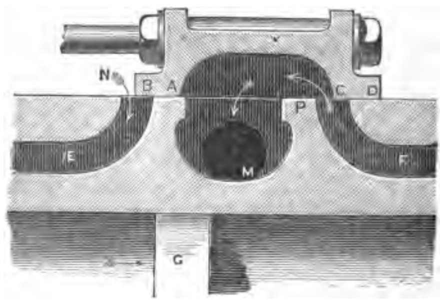


FIG. 17.

a certain part of the stroke, steam would enter on both sides of the piston. Let us examine the valve (Fig. 17). Now, in the normal valve, the flanges B A, C D at the middle of the stroke of the eccentric just covered the steam-ports; at this time the piston was ready to begin its stroke. As the valve moved from N to B, steam was admitted, and forced the piston G to the end of the cylinder, the steam behind G escaping through F to the condenser by the exhaust M. The valve uncovered the ports E and F in this way, and had just covered

them again, when G had gone as far to the right as possible. The valve is now moving to the left, and uncovers the ports again, F receiving steam, and E allowing steam to escape to M as the piston begins to move to the left. Here one end of the cylinder is open to the condenser, and the other to the boiler during the whole stroke. It was found that with this normal valve there was too little time for the steam to escape by the exhaust-port, so that the back-pressure was considerable; and, besides, there was no expansion.

Lead.—*By changing the position of the eccentric on the crank-shaft, the steam-port may be closed and the exhaust-port opened before the end of the stroke.* Still, there is no expansion allowed, for the exhaust is opened as the steam ceases to enter. This is called giving *lead* to the valve, the amount of lead being the distance which the valve has reached beyond its middle position at the beginning of the stroke of the piston. We shall call this the *advance* of the valve, to distinguish it from the *lead of the outside edge of the valve*, which is the amount by which the port is open at the beginning of the stroke. In future, when *lead* is spoken of, the amount of opening of the port at the beginning of stroke must be understood.

Lead may be measured (1) in inches, or (2) as the angle through which the eccentric has to turn to produce the lead, or (3) as a fraction of the half-throw of the valve.

Lap.—*It is found that the steam-port will close some time before the exhaust is opened, if we make the distance* B D *longer by adding to the breadth of the flanges* B A *and* C D.

When the valve is at half-stroke, as in A (Fig. 18), the distance B H, or F D, is called the *lap*, or the *outside lap*; the distance G A, or P E, being *inside lap*. *Negative* inside lap is called *inside clearance.*

Lap, like lead, may be measured (1) in inches ; (2) as the angle through which the eccentric has to turn as the valve moves through the distance from B to H ; or (3) as a fraction of the half-throw of the valve.

Fig. 18 gives three views of the three-ported, or locomotive slide. *b* represents the configuration at the beginning of the stroke of the piston, B′ H being the

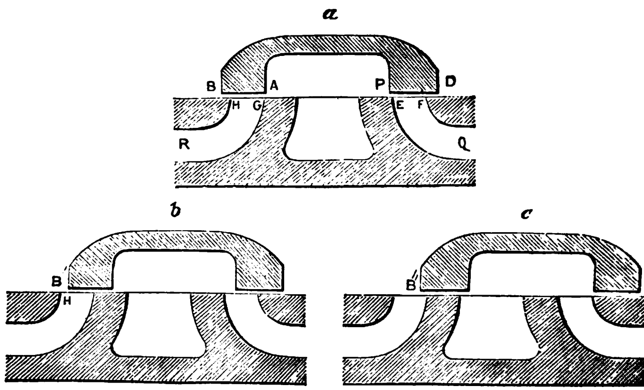


FIG. 18.

lead. Evidently, in all cases, B′ H the lead + the lap = advance of the valve. *c* represents the end of the valve-stroke. The valve afterwards assumes its position in *b*, and as it approaches that in *a* the steam-port is closed. Expansion is now taking place in the cylinder ; and when the valve moves over a distance equal to the *outside* lap, the position shown in *a* is regained.

The further motion to the left of A and B will now be represented by the motion which P and D had to the right during the preceding operations.

As the valve moves through a distance equal to the *inside* lap, the exhaust-port opens; so that from the closing of the steam-port till the exhaust-port opens, the valve has moved over a distance equal to the sum of the inside and outside laps.

The exhaust-port is well opened at the end of the stroke of the piston, as may be seen at $E^1 P^1$ in *b*. In the back-stroke *c*, the exhaust-port is quite open, being

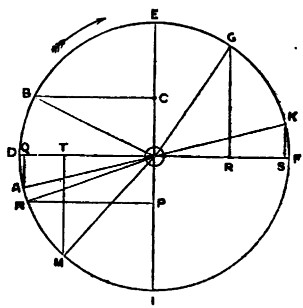


FIG. 19.

closed as the valve moves into the position *b*. When P^1 reaches E^1 the exhaust is completely closed, and cushioning begins. It will be seen, that after cushioning begins, the valve has to travel through a distance equal to the *inside lap* before it again reaches the middle position.

Neglecting the angularities of the connecting and eccentric rods, let D F (Fig. 19) represent the diameter of the circular path of the crank, and let the feet of perpendiculars from dif-

ferent points of the curve on the line D F represent positions of the piston at different times.

Let us consider the forward-stroke D F, in which the crank moves round D E F.

Again, let I E *on another scale* represent the stroke of the valve, so that the circle may represent either the path of the crank-pin or the path of the centre of the eccentric disc. In the latter case, I is the beginning, and E the end of the stroke.

Lead, lap, &c., will be measured according to the second scale, in which I E represents the stroke of the valve.

In the normal valve, when the crank-pin and piston are at D, the centres of the eccentric and valve are at D and O.

Thus the eccentric is in mid-stroke at the beginning of the stroke of the piston.

Let the throw of the valve E I be given, as well as the positions T and Q of the piston in the back-stroke, indicating times when the exhaust-port closes and the steam-port opens.

Let R be the position of the piston when steam is cut off; that is, let the length D R of the whole stroke D F be made when steam is cut off.

To find the lead, the advance of the valve, and the outside and inside lap, draw perpendiculars Q A and R G.

The crank moves over the space A D during the time of admission before the stroke begins, the piston moves over Q D, but the valve moves over A Q, which, therefore, represents the *lead*.

Steam is being admitted during the time in which the crank-pin moves from A to G. It is evident that the eccentric will be at E when half this time has elapsed, as E represents the end of the stroke (consult *c*, Fig. 18); so that if E B = $\frac{1}{2}$ A G, the *arc* A B (rather, the projection of the arc on E I), *or the angle* A O B, *represents the advance of the valve*, which is made up of the lap and lead; and as the outside lap is always moved through from the middle position of the valve, the *distance* O C *is the outside lap*.

Again, let T be the position of the piston in the back-stroke when the exhaust-port closes. Draw the perpendicular T M, make M N equal to the advance A B, draw N P parallel to D F, *then* O P *represents the inside lap.*

Make G K equal to the sum of the arcs B D and D N, which correspond to the outside and inside laps respectively; drop the perpendicular K S; then will K and S *represent the position of the crank and piston when the exhaust-port opens.*

The results found above by construction may easily be determined by calculation.

Let the stroke of the piston D F be r' times D R, q times D Q, and r'' times D T. Let a be the angular advance of the eccentric, or A O B, let b' be the angle D O B, corresponding to the outside lap, b'' being the angle D O N, corresponding to the inside lap. Then

$$a = \frac{\cos^{-1}\left(\frac{2}{r'} - 1\right) + \cos^{-1}\left(1 - \frac{2}{q}\right)}{2} \qquad (1)$$

$$b' = \frac{\cos^{-1}\left(\frac{2}{r'} - 1\right) - \cos^{-1}\left(1 - \frac{2}{q}\right)}{2} \qquad (2)$$

$$b'' = \cos^{-1}\left(1 - \frac{2}{r''}\right) - a \qquad (3)$$

And outside lap $=$ O E. $\sin b'$ (4)

inside lap $=$ O E. $\sin b''$ (5)

and it will be found that for release

$$\frac{\text{D S}}{\text{D F}} = \frac{1 + \cos(a - b'')}{2} \qquad (6)$$

Again, given a and the inside and the outside laps, and, consequently, given b' and b'', it will be seen that—

$$\frac{2}{q} = 1 - \cos(a - b') \qquad (7)$$

$$\frac{2}{r'} = 1 + \cos(a + b') \qquad (8)$$

$$\frac{1}{r''} = 1 - \cos(a + b'') \qquad (9)$$

which enable us to find the periods of admission, cut-off, and compression respectively.

From (4), (5), (7), (8), and (9), it will be seen that as the travel of a valve gets smaller, the lap and lead remaining as before, the steam is cut off at shorter and shorter periods of the stroke, is released and compressed more towards the ends of the stroke, and is admitted at greater and greater periods before the end of the back-stroke.

91. For practical purposes, the following rules will be found useful. Given lap, lead, and travel of valve :—

(1.) $\text{Sin (angular advance)} = \frac{\text{lap} + \text{lead}}{\text{travel}}.$

(2.) To find the cut-off.—Find the obtuse angle whose sine is *lap*. From this angle subtract the angular advance found above. If the remaining angle is obtuse, add *unity* to its cosine, which is negative. If acute, subtract it from *unity*. Half this will give the cut-off as a fraction of the whole stroke.

(3.) To find the period of the back-stroke at which steam is admitted.—Find the *acute angle* whose sine is $\frac{lap}{\text{half-travel}}$; subtract this angle from the angular advance found by (1); find the cosine of the difference, and subtract it from one; half this difference will be the answer.

(4.) Period of stroke at which release takes place

$$= \frac{1 + \cos\text{ (angular advance)}}{2}.$$

92. The following table will show the effects of changes in the travel, lap, and lead of the valve of a

locomotive engine whose cylinder was 15″ in diameter, stroke 22 inches long, and ports $1\frac{1}{4}$ inches wide.

$\frac{1}{r'}$ = the fraction of back-stroke performed after admission.

$\frac{1}{q}$ = the fraction of the stroke performed when the steam was cut off.

$\frac{1}{r''}$ = the fraction of the back-stroke performed before the exhaust closed; this is also equal to the fraction of the forward-stroke performed before release.

	I. Travel decreasing. Lap 1 inch, Lead $\frac{5}{16}$.					II. Lap increasing. Travel $4\frac{1}{2}$, Lead $\frac{5}{16}$.					III. Lead increasing. Travel $4\frac{1}{2}$, Lap 1 inch.				
	Travel in inches.					Lap.					Lead.				
	$5\frac{1}{4}$	$4\frac{1}{2}$	$3\frac{3}{4}$	3	$2\frac{5}{8}$	0	$\frac{1}{2}$	1	$1\frac{1}{2}$	$1\frac{15}{16}$	0	$\frac{5}{16}$	$\frac{10}{16}$	$\frac{15}{16}$	$\frac{20}{16}$
$\frac{1}{r'}$	·004	·006	·011	·026	·122	·007	·009	·006	·011	·065	0	·006	·030	·085	·280
$\frac{1}{q}$	·815	·735	·600	·390	·120	·992	·915	·735	·490	·065	·805	·735	·640	·535	·280
$\frac{1}{r''}$	·935	·910	·860	·735	·500	·992	·970	·910	·810	·500	·95	·91	·84	75	·50

From this table it is seen, that by varying the travel of the valve we may get a great amount of expansion in the cylinder.

More benefit arises, however, from varying the lap; partly from the greater amount of expansion allowed; partly from the periods of pre-admission being shorter.

Where much variation in expansion is necessary, the link-motion (*see* Book III.), which varies *travel*, will be

K 2

found most efficient, contrivances for varying lap being rather cumbrous.

Inside clearance has but little effect on the distribution of steam. It hastens the release, and hence, also, delays the compression. Inside lap is also unimportant; it delays the release, and hastens the compression. Of the two, inside clearance is the less hurtful.

It is to be remarked, that from the angularity of the connecting-rod, with equal lap at both ends of the valves, steam is cut off sooner in the back than the front stroke. This is remedied by increasing the front lap. The periods of release are approximated in the same way.

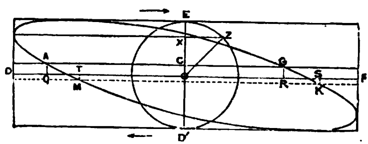


FIG. 20.

Using the same scale for all dimensions, let D F (Fig. 20) represent the stroke of the piston, and E I, bisected at O and bisecting D F at right angles, the stroke of the valve. Describe circles on these lines as diameters; the circles represent the paths of the crank-pin (not shown) and centre of the eccentric. Make the angle E O Z equal to the angle of advance of the valve found in last article. Then the eccentric centre is at Z when the crank-pin is at D. Divide both circles into the same number of equal parts, drawing perpendiculars, in the one case to D F, and in the other to E I. Note corresponding points, and draw from them lines parallel to E I and D F. The points in which corresponding lines meet lie in an elliptic curve. Let R, Q, and T be taken as in Art. 92, then A Q will be found equal to R G, and will be the outside lap; M T will be the inside lap; and if M K be drawn parallel to D F, and K S be drawn, S will represent the position of the piston at the point of release.

Ordinates of the curve parallel to E I denote the motion of the valve, those parallel to D F denote the motion of the piston.

Thus, the part of the curve above A G represents the motions of the piston and valve during the admission of steam; the part G K, during expansion before release; the part K M, during exhaust; the part M A, during compression.

When the angularities of the connecting and eccentric rods are also considered, it is to be remembered that the piston or eccentric moves faster at some places, and at others slower, than the feet of the ordinates drawn from equidistant points in the circumference of a circle to its diameter.

After finding the angle of advance of the valve, by the method described in last article, a skeleton drawing must now be used to determine the corresponding points in E I and DF, and the curve (which will no longer be elliptical) drawn as before.

93. The *diagrammeagraphe* of M. Pichault is described in *Engineering*, of the 15th December, 1871. By it we may obtain from the engine a diagram of the relative positions of the crank and slide-valve and ports at any instant. A disc of paper on the end of the crank-shaft turns with the shaft, its centre being in the axis. Along a diameter of this disc a slide moves, holding a pencil. The slide has a motion which it gets from the valve, and which corresponds exactly to that of the valve. A pencil on the slide will describe on the paper a curve, whose radii to different points from the centre of the disc will determine the motion of the valve. The valve and slide are fixed in the position in which they happen to be when the steam-port is just closed or about to be opened, and a circle described on the disc by making the engine revolve. This circle cuts the former curve in points, the radii to which show the positions of the crank when steam is admitted or cut off. Readers are referred to *Engineering* for a detailed account of the arrangement.

94. **Long D Slide Valve.**—In principle this is almost identical with the locomotive slide. Steam enters the ports from the middle space L (Fig. 21), and leaves during exhaust for the spaces K or G, which communicate through the tubular valve, and are in connection with the condenser.

Steam from the boiler surrounds the middle part of the valve C D. Lap is now inside, instead of being outside. The surfaces A E and B F of the valve are well packed and move steam-tight in the chest.

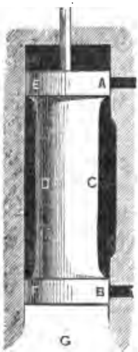


FIG. 21.

It is evident, that with this long valve the passages to the cylinder may be made very short. There is generally a loss in engines with long passages, much of the steam filling the passages, being wasted at every stroke; but it will be shown presently that with a proper arrangement of valve-motion there need be no loss whatever.

95. **The Short D Valve** consists of two pistons, shaped on their circumferences like A E and B F (Fig. 21), fixed firmly at a distance apart on a rod. The space between them is always filled with steam, and there are two exhaust-spaces, K and G, *separately* connected with the condenser.

96. **Separate Slide Valves for Expansion.**—Of these there have been many forms. In general they are cumbrous, and give rise to much friction.

In *Fenton's*, a spiral feather on the crank-shaft allowed the eccentric to turn independently of the shaft at a certain part of the stroke.

Edwards had a second valve sliding on the back of the first, which was constructed with two additional steam-passages. This shifted its position at certain times by meeting tappets fixed on the valve-chest.

Bodmer and Meyer used blocks moved from the cross-head, instead of the secondary valve of Edwards.

Hawthorn employed a rectangular frame surrounding the ordinary valve, and capable of sliding steam-tight over the top of the flange at each end.

Gonzenbach employed two slide-valves, one of which admitted steam into the valve-chest of the other.

A rectangular frame, which acts as a steam-valve, and surrounds an ordinary slide, regulating the exhaust and worked

by a tappet, whilst the exhaust is worked by an eccentric, forms a complicated valve-motion, still to be seen in some horizontal stationary engines, and in many marine-engines (*see* Fig. 22).

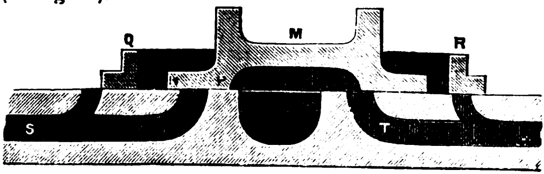


FIG. 22.

97. **Cams and Tappets.**—When a shaft is revolving at a regular rate, and it is necessary to obtain a *continuous* reciprocating motion, which shall be the same at the same period during every revolution, we employ a crank and connecting-rod, or an eccentric. When, however, as in valve-motions, it is necessary to get a discontinuous reciprocating motion, we employ other cams, or tappets. The eccentric in different forms, the swash-plate, the revolving heart touched on both sides by two points on a reciprocating bar, and other cams, are fully described in many books on Mechanism.

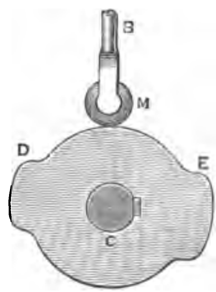


FIG. 23.

Fig. 23 shows a cam or tappet, commonly employed to open steam-valves. E D is the solid cam keyed to a revolving shaft C. At the end of the valve-rod B is the friction-roller M, which merely turns on its axis during the greater part of a revolution, but which is lifted at certain times, and let fall again by means of the raised parts E and D. By moving E along the shaft C, different arrangements of raised parts may be employed, to give motion to the valve; and when this is

possible—that is, when there is a feather on the shaft, and no key is used, and when the tappet is placed in connection with the governor—there is no need for a throttle-valve, for the governor will now give much or little expansion, as the load is small or great.

Tappet Motion for Beam-engines.—The arrangement of valves most common in stationary beam-engines is that shown roughly in Fig. 24. J K Q and M L N are two valve-casings connected by means of two hollow columns, one of which is shown in P. One of these columns is always open to the condenser; the other may be connected with the boiler by means of a cock. J, K, and Q are three compartments in the upper casing; M, L, and N in the lower. Let us suppose that P is the exhaust-column, connecting the compartments Q and N, while a steam-column always connects M and J with the boiler.

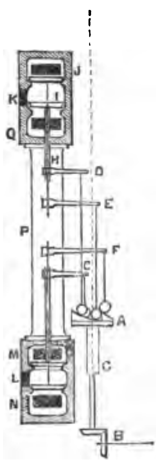


FIG. 24.

The casings are firmly fastened by screws at L and K to flanges on the upper and lower ports of the cylinder, so that by means of the openings the middle compartments K and L are always in communication with opposite sides of the piston.

Four valves are shown: these must be understood to be double-beat valves, although represented in the drawing as of the ordinary conical kind. All the valves open upwards, and are worked by means of two solid and two hollow *rods*, the latter allowing the former to pass through them steam-tight. The four *rods* have four arms securely fastened to them; and these, again, have long vertical rods fastened to their other extremities, terminating in friction-rollers, which are almost on the same level when the valves are closed. When, therefore, one of the friction-rollers is lifted, a valve is opened which cannot be closed until the roller falls.

A is a horizontal plate carrying the two tappets. The smaller of these is called the *steam-tappet*, as, by means of the inner friction-rollers, it works the steam-valves at J and M. The other, which is higher and much longer, by means of the outside friction-rollers, works the exhaust-valves at Q and N.

When part of the load is suddenly thrown off an engine, it is convenient that a change should at once be made on the period at which the steam is cut off, so that less steam may enter the cylinder.

In ordinary engines, provided with a slide-valve, the throttle-valve is worked from the governor, so that the steam is wire-drawn.

Changes in the period of cut-off afford a much better means, however, of altering the steam supply, and, in the present case, these are made by placing the steam-tappet at different distances from its axis of revolution. There are many contrivances for allowing the governor to produce changes in the position of the tappet, of which the following seems best.

The steam-tappet is made to move radially by means of a sliding rack, whose teeth are placed downwards, and fit the teeth of a small spur-pinion, which is supported from two little brackets on the under-side of the plate. Whenever the pinion is turned, the tappet moves out or in.

The pinion also works into a rack, whose teeth appear through a vertical slot in the shaft C. This rack turns with the shaft, and is jointed to the end of the governor-rod which comes down from above; so that, whenever motion is given to the rod, the rack is lifted, the pinion is turned, and the horizontal rack moves the tappet.

CHAPTER VII.

CONDENSER, AIR-PUMP, AND FEED-PUMP.

98. Fig. 25 shows the connection which these parts of the engine have with each other. Steam enters the condenser C by Q, and, meeting injection-water from B, falls in the liquid state, heating the injection-water during its condensation. When the bucket D is lifted, the heated water passes through the *foot-valve* P, and is, from the nature of the valve, prevented from again entering the condenser; so that it eventually passes through the bucket and the delivery-valve F into the hot-well E.

When the feed-pump plunger G is lifted, water passes upwards through the valve H. When G is forced downwards, the water is pressed through the valve I and the pipe J, either to the boiler or to a cistern.

The Condenser may have any convenient shape. Its capacity is from $\frac{1}{4}$ to $\frac{1}{2}$ that of the cylinder. The area of the injection-orifice is about $\frac{1}{250}$th of that of the piston in ordinary engines—a formula which allows $\frac{1}{16}$th of a square inch of injection-orifice per cubic foot of water evaporated per hour.

From the calculated quantity of injection-water required per cubic foot of steam evaporated, given in formula (7) at the end of the book, the proper area of orifice may be calculated, and is found to be about half that practically employed.

Sometimes, when pure water is scarce, ***surface-condensation*** is employed. Here the steam passes into a number of tubes, and is pumped from a vessel connecting their lower extremities by means of a small air-pump. The best effects of surface-condensation were obtained by Joule, who passed the condensing-water through pipes, each of which surrounded a copper steam-tube. The water flowed in a direction opposite to that of the steam-current. He found it possible to condense 100 lb. of steam per hour per square foot of tube surface.

In some experiments on marine-engines, using surface-condensation, it was found that three to four pounds of steam per hour were condensed per square foot of tube-surface; the pressure of uncondensed steam and air being

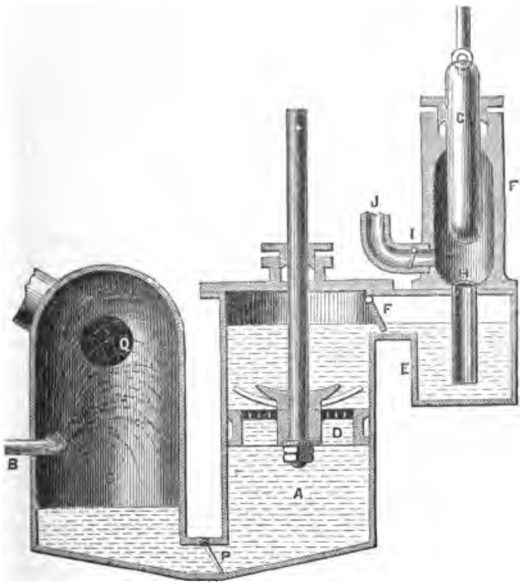


FIG. 25.

1·7 lb. per square inch. Perhaps the best results are obtained when the exhaust-steam is first subjected to surface-condensation, the change in state being completed with the help of a small injection-jet.

With surface-condensation there is no great expenditure of water; so that this may be very pure when it is first put into the boiler, and may be kept pure by the place of that which leaks away being supplied from separate distillation. Sometimes the condensing-water is also dispensed with, the surface condenser being formed of a great number of tubes revolving rapidly in the air.

From the nature of the calculations at the end of the book on the efficiency of engines, the necessity is seen of keeping the condenser at a low temperature. Even when very hot feed-water is required, it is better to heat it during its passage to the boiler, than to have it hot leaving the condenser.

It may be shown, that when the condenser is kept at 38° C. the ratio of the amount of effective condensation to the amount of water lifted by the air-pump is a maximum.

From all condensers the water must be pumped away, and it is necessary for the pump to overcome the pressure of the atmosphere. Now, when the condenser is elevated 33 feet above the ordinary level, and when the water of condensation falls into a long pipe, the lower opening of which is beneath the surface of water in a cistern, a column will always be maintained in the pipe by atmospheric pressure, and the condensed water will escape at the bottom into the cistern. At first sight it may seem, that with this arrangement there is no loss as in an air-pump; but a little consideration will show that the engine still does work in removing the condensed water. In fact, the steam and injection-water have to be raised to a height of 33 feet; and in doing this, whether by creating a vacuum or otherwise, an amount of work is done which is equivalent to that done by an ordinary air-pump. In the *jet-condenser*, also, the air-pump is dispensed with. The velocity with which steam, even when at a low pressure, enters a vacuum, is taken advantage of to convey the water of condensation into the hot-well. The central jet of injection-water is surrounded by a nozzle for exhaust-steam; and the receiving-pipe gradually expands towards the hot-well. The principle of this condenser will be better understood when Giffard's injector is known.

Bourne has designed a condenser, in which a small jet of injection-water meets the descending steam, and falls almost

at the temperature of 100° into a suitable vessel, from which it may be pumped to supply the boiler. A second jet of water meets the uncondensed steam lower down, completing the action of the first. By this arrangement there is less injection necessary, and the water in the marine-boiler, to which it is applied, need not remain saturated with salt to such a great extent as is usual.

The area of the *foot-valve* varies from $\frac{1}{3}$rd to $\frac{7}{8}$th, or in pumps whose buckets move very fast, to the full size of the area of the opening in the bucket.

The valve is now usually of india-rubber (Art. 75). In marine-engines the seat sometimes extends over the pump-bottom.

99. **The Air-pump,** when single-acting, has a capacity $\frac{1}{8}$th or $\frac{1}{6}$th of that of the cylinder. The resistance to the motion of the bucket may be measured by a back-pressure in the steam-cylinder of $\frac{1}{2}$ lb. or $\frac{1}{4}$ lb. per square inch.

The pump is often made of brass, with brass valve castings, but it is usually a cast-iron pump which has been lined with copper after being bored. The rod is of Muntz metal, or of wrought iron covered with brass. In this latter case, water must never be allowed to get between the brass and iron (*see* Art. 54), and brass fastenings, keys or nuts, are needed. The pump-rod, like that of the piston, is attached in various ways. These may be understood from Art. 102.

The bucket is packed with hemp, like the piston of Art. 101, a junc-ring being necessary. Sometimes the bucket is turned accurately to fit the pump with one or more grooves on its rim. These hold water, and give sufficient water-tightness.

The valve of the *air-pump bucket* is of india-rubber, the brass seat being perforated with a great number of small orifices. In many cases the *delivery-valve* is the same as that in the bucket, and extends over the top of the pump. When this is the case, that there

may be no difficulty in reaching the bucket, the delivery-valve is attached to the cover, the pump-rod passing through a tube connecting both.

Maudslay employed the divided conical valve of Art. 77 for his bucket; and when it was well madc, he found that it gave satisfactory results.

100. **Double-acting Air Pump.**—The principle of the double-acting pump is shown in the rough sketch (Fig. 26) in which P represents a solid piston

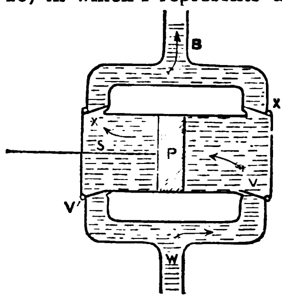


FIG. 26.

working in a pump provided with four sets of valves. These valves are usually of india-rubber (*see* Art. 75). The piston is moving to the left, water follows it from W, filling the space behind it through the valve V, X being closed, and other water is pressed from before it, passing through X^1. When P makes the return-stroke, the valves X^1 and V will close, and V^1 and X will open, so that water may still pass up into B, coming from W.

This pump may be connected with the condenser in many ways; V and V^1 may merely take the place of the foot-valve, or another valve may be employed at the extremity of W.

It is always found that these double-acting pumps work irregularly when much clearance is left at the

ends of the stroke, principally from the lodgment of air, which had been dissolved in the water.

CHAPTER VIII.

THE PISTON.

101. **The Piston.**—When the packing is of hemp, the body of the piston is somewhat like that shown in section at C, in Fig. 27, the ring A being away. Well-

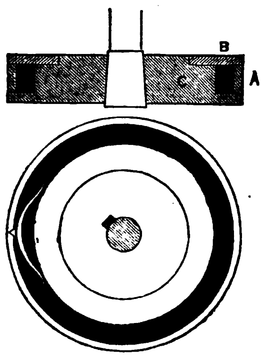


FIG. 27.

plaited rope-yarn and oakum, well saturated with tallow, are beaten into tight layers in the space A, before the junc-ring B is screwed down. In the figure, metallic packing is used instead of hemp. A cast-iron ring cut at D, and shaped there to receive a wedge pressed forward by a strong spring, is always tending to get larger, fitting the cylinder all round as closely

as possible. This ring is carefully scraped, so as to be steam-tight with the body of the piston below A and the junc-ring B, and is kept from sliding by means of a pin fixed in C.

It is not of much consequence whether one or many springs be employed for a ring like this. With one there is a tendency to wear the cylinder oval; on the other hand, the complexity of several springs is undesirable.

Fig. 28 shows a ring much used in locomotive pistons, the spring having a bearing all round the inside of the ring, and seeming to act like a number of small springs.

A cast-iron ring turned a little larger than the cylinder, with a small piece cut out where the ring is thinnest, if it gradually varies in thickness, may be compressed to the proper size, and again turned. when it will be found to exert sufficient pressure all round its circumference to render auxiliary springs unnecessary.

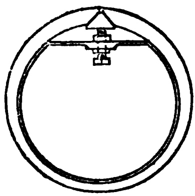


FIG. 28.

The cut of such a ring as this —indeed, the cut of any piston-ring—ought to be oblique to the direction of motion of the piston, and never straight across; and the joint ought to have a tongue-piece fastened to one side, and flush with the surface, so that there may be no ridges or grooves allowed to form.

With two or more rings, the joints ought to be broken, or placed 90° apart for the same reason, and also that there may be less oval-wearing of the piston.

Instead of springs and wedges, tight packings of hemp are often made behind these rings, which are, in such cases, sharpened on the upper edge by being cut away from the inside.

The body of the piston is solid in small pistons, as in Fig. 27, and hollow in pistons of large diameter. In locomotive-engines it is often forged on the rod. Sometimes, instead of the inside part of the body being cylindric, it consists of three or more horns, or thick ribs, on a bottom plate. Again, small pistons may consist merely of a thick disc, in which two or more grooves have been turned to receive small packing rings.

Piston-springs are seldom made strong enough. Their shape and position seem immaterial. The following examples of piston-packing may be mentioned :—

Seaward.—Marine-engine piston : One ring, cut at one place, the cut being covered with a segmental piece on the outside, turned up with the ring. Six springs, like that in Fig. 27, force the ring outwards all round.

Napier.—Marine-engine piston much in use: One ring cut in many places. A flat block, used instead of a wedge at every cut, is pressed out by means of a vertical spring, the back of which fits a groove in the body of the piston.

Fawcett.—One ring with four cuts : Four blocks, as in Napier's piston, are pressed outwards by means of ordinary springs.

Girdwood.—Several narrow, small-grooved rings cut and bent to fit the piston. Each of these has a narrow groove. A hemp-packing often used behind.

Molineux and *Nicholl.*—Rings forced out by the pressure of steam.

Forrester.—Locomotive piston : One ring divided into four parts. Hence there are four wedges and four springs, which rest on four horns on the body of the piston.

Dircks.—One ring with one cut and one wedge. One circular spring encircling four horns on the body, and attached to three other parts of the ring by means of screws.

Bury.—Two rings, each with four wedges, no two cuts being over each other. Each spring has its centre caught by a screw from the body, its two ends resting on wedges in the upper and lower ring respectively.

Stephenson.—Locomotive-piston : Three concentric rings, each cut at three places. The ends of three springs press on

the rings, their middle points being caught by screws from the body.

102. **The Piston-rod.**—This is now usually of steel, or of case-hardened wrought iron, its diameter varying from $\frac{1}{10}$th to $\frac{1}{7}$th of the diameter of the piston. It is turned very truly, and may be fastened to the piston in many ways. Fig. 27 represents the common arrangement in large engines, the tapered end of the rod sometimes being fastened by means of a key and cutter; sometimes being screwed at the top, and held by means of a round, or hexagonal nut.

Small grooves turned out of the rod, just below and above the piston, may be packed with hemp.

Instead of one, several piston-rods are often employed in large marine-engines; or, to save space, a trunk may be substituted, enabling a connecting-rod to be fastened directly to the piston.

CHAPTER IX.

PARALLEL MOTION.

103. IN the diagram of the beam-engine (Fig. 14), J H I F G is the parallel motion, or, rather, that part of the parallel motion is shown which is on this side of the beam. J H is a *radius bar*, which causes the point H to move in the arc of a circle with J as centre; H I and F G are the *back-link* and *main-link*, whose lengths are usually that of the crank; G H is a *parallel bar*. These links and bars hide from our view corresponding and similar links and bars on the other side of the beam.

The piston-rod is keyed at G into a round hole in the middle of the *cross-head*, or it may be screwed at the end and fastened by means of a nut, as in many locomotive-engines. Sometimes, in marine-engines, both the nut and keys are provided.

The *main-links* connect the outer bearing of the *cross-head* with the end-stud of the beam. Each of the main-links has a cross-section which is one-half that of the piston-rod, that together they may transmit as great a pull, or thrust, as the piston-rod, without breaking.

Two inner bearings on the cross-head are caught by the ends of the parallel bars, which are bushed in the manner described in Art. 112. The other ends of the parallel bars are provided with screws, and may be fastened into screwed holes *in the cross-bar of the back-links*. This cross-bar has four bearings, two on the outside for the ends of the radius bars, those on the inside being intended for the back-links. The *radius bars* are short connecting-rods, both of whose ends are bushed, like the single end of a parallel bar in the manner described in Art. 112. They connect the bearings on the back-link with studs on the spring-beams. Perhaps it is better to employ a shaft, working in plummer blocks for the ends of the radius bars, as this will allow of adjustment. The lengths of the parallel bars ought to be adjustable.

The back-links have three sets of brasses each, as may be seen from the general drawing of the beam-engine, one of these being intended for a stud in the beam, one for the *cross-head of the air-pump rod*, and the third for the cross-bar. The air-pump cross-head is often bent downwards in the middle, as the space overhead is very confined.

In practice, in mill-engines, it is usual to have the radius and parallel bars of the same length; and from what follows it will be seen that they must each, in consequence, be equal to half the length of the beam.

In this case, the studs for the ends of the radius bars are in one plane with the centre line of the cylinder.

104. In Fig. 29, let A be the centre of the beam ; B C the link to which the air-pump rod is attached ; M the point of attachment ; and C D the radius bar. Let B C be at right angles to A B and C D. Suppose A B and D C to be moving in the direction indicated by the

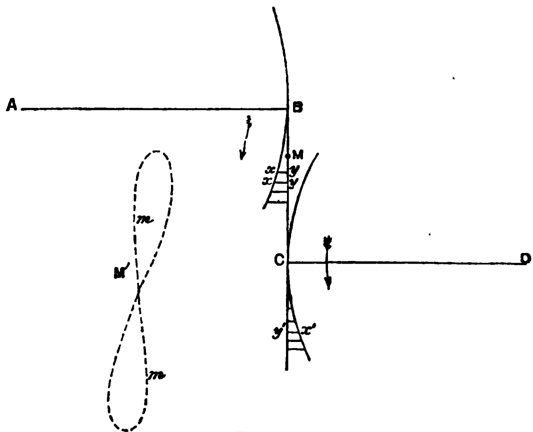


FIG. 29.

arrows, and let us consider the different positions on the arcs of the ends of the link B C. B is travelling to the left, whilst C moves to the right ; thus there must be some point M on the link B C, which neither moves to the left nor right, and that point may be found by a mathematical calculation.

Watt very clearly stated the principle of the parallel motion when he said that the points B C described

arcs whose convexities lay in different directions, and compensated for each other's variation from a straight line; so that a certain point M at the top of the pump-rod, lying between these convexities, ascended or descended in a straight line.

Now, if A B is greater than D C, the little distances xy, by which the point B is drawn to the left, are less than the corresponding distances x' y'; that is, C is drawn to the right more than B to the left, and hence the point M must be nearer B than C. In fact, we shall show that BM : MC :: DC : AB, or that the link BC is divided into segments, which are inversely proportional to the lengths of the nearest vibrating rods; and that, when we place the link in all possible positions, the point M traces out the curve M′ in space, the part of it, $m\ m$, being Watt's straight, or nearly straight, line. Every point in BC would, in such a case, trace out a somewhat similar curve to M′; but this is the curve which most approximates to a straight line at any place.

Let A B make the angle θ with its present position, and C D the angle ϕ, both being positive, and both being expressed in circular measure. (I.) Let us find the position of M, for which the path is most nearly a vertical line when the bars begin to move.

Evidently $\frac{\text{BM}}{\text{CM}} = \frac{\text{AB}\,(1 - \cos\theta)}{\text{CD}\,(1 - \cos\phi)} = \frac{\text{AB}.\ \theta^2}{\text{CD}.\ \phi^2}$, since θ and ϕ are very small.

Again, the small arc travelled through by B is almost equal to that travelled through by C; hence—

$$\text{B}.\ \theta = \text{CD}.\ \phi \therefore \frac{\theta}{\phi} = \frac{\text{CD}}{\text{AB}} \text{ and } \frac{\text{AB}.\ \theta^2}{\text{CD}.\ \phi^2} = \frac{\text{AB}.}{\text{CD}}\ \frac{\text{CD}^2}{\text{AB}^2} = \frac{\text{CD}}{\text{AB}}$$

Hence

$$\frac{\text{BM}}{\text{CM}} = \frac{\text{CD}}{\text{AB}}.$$

And thus the position of M is determined.

Let α be the angle made by the link BC with its present position—that is, with a vertical line at any time. Then, if B is the origin chosen for co-ordinates, it may be shown that

$$\text{AB} \cos \theta + \text{BM} \sin \alpha - x = \text{AB} \quad (1)$$
$$\text{AB} \sin \theta + \text{BM} \cos \alpha = -y \quad (2)$$
$$\text{AB} \cos \theta + \text{BC}. \sin \alpha + \text{CD} \cos \phi = \text{AB} + \text{CD} \quad (3)$$
$$\text{AB} \sin \theta + \text{BC} \cos \alpha = \text{CD} \sin \phi + \text{BC} \quad (4)$$

From these equations eliminating θ, α and ϕ, we may get the equation to the curve traced out by M.

105. Again, it may be seen from these equations that α, or the deviation of M from the vertical,

$$= \frac{\text{CD}^2 (1 - \cos \phi) \sim \text{AB}^2 (1 - \cos \theta)}{\text{CD} + \text{AB}} \dots (5)$$

In many ordinary cases, CD is made equal to AB. In this case the deviation of M from the vertical

$$= \frac{\text{CD}}{2} (\cos \theta \sim \cos \phi). \dots (5)^*$$

To use this formula (5), or (5)*, assume a value for θ. Eliminate α from the equations 3 and 4, and thus find ϕ in terms of θ. To simplify these expressions, find α from (3) approximately, that is by assuming $\alpha = \sin \alpha$, and also that $\cos \theta = \cos \phi$, hence

$$\alpha = \frac{\text{AB} + \text{CD}}{\text{BC}} (1 - \cos \theta) \dots (6)$$

which gives with (4) the value of ϕ. And thus the expression (5) or (5)* may be calculated.

Example.—$\theta = 15^\circ$. Let $\text{AB} = a$ and $\text{CD} = \frac{3}{6}a$, and $\text{BC} = \frac{1}{8}a$. Find the deviation of M from the vertical at

the end of the stroke, M being taken in the regular way—that is,

$$\frac{BM}{CM} = \frac{CD}{AB} = \tfrac{8}{5}.$$

From (6), a in circular measure $= \frac{24}{5}(1 - {}^{\cdot}965926)$ $= {}^{\cdot}1635$, or $a = 9^\circ 25'$, and from (4) ${}^{\cdot}259\,a + \frac{1}{3} a \times {}^{\cdot}988$ $= \frac{8}{5} \sin \phi + \frac{1}{3} a$; hence $\sin \phi = {}^{\cdot}425$ $\therefore \phi = 25^\circ\ 10'$; hence from (5) the deviation of M from the vertical

$$= \frac{\frac{9a^2}{25}(1 - {}^{\cdot}9063) \backsim a^2 (1 - {}^{\cdot}966)}{\frac{8a}{5}}$$

$$= {}^{\cdot}00017\, a.$$

106. When the centres D and A are on the same side of the line B C, M will be found in B C produced, but in every case $\frac{BM}{CM} = \frac{CD}{AB}$

This form of parallel motion is never used in stationary-engines, but its use is often convenient in marine-engines.

107. In Fig. 30, A B C D is as before, but A B is now continued to ; E and two jointed bars, C F and E F, are added in such a way that the straight line A M produced passes through F. Hence

$$\frac{AE}{AB} = \frac{EF}{BM} = \frac{BC}{BM} = \frac{AB + CD}{CD}.$$

Now, the figure B C F E is always a parallelogram—that is, the line E F is always parallel to B M ; and it may be shown that, whenever this is the case, the points M and F must trace out similar paths ; hence F traces out a curve similar to M′ (Fig. 29), so that, for small angular

displacements of the arms, the path of F is sensibly a straight line. The deviation of F from the vertical

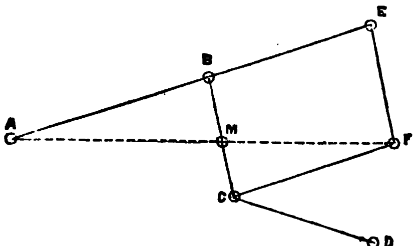


FIG. 30.

may readily be calculated, as it is equal to the deviation of M multiplied by $\frac{AF}{AM}$, or by $\frac{BC}{BM}$, or by $\frac{AB + CD}{CD}$. From (5) it is equal to

$$\frac{CD^2\,(1 - \cos\phi) \backsim AB^2\,(1 - \cos\theta)}{CD},$$

which becomes $CD\,(\cos\theta \backsim \cos\phi)$, when $AB = CD$.

108. Produce the line D M till it meets a line drawn through B parallel to C D. Let this point be F′, then F′, B, and C are the three corners of a new jointed parallelogram, which may be employed to enable F′ to move in an approximate straight line. This form of parallel motion is common in marine-engines.

D and A being on the same side of B C, we find that another pair of parallel motions are possible by producing the lines D M and A M respectively.

These four different parallel motions again take other distinct forms, as A B is greater than, equal to, or less than C D.

109. In the Gorgon Engines, a parallel motion was introduced, the principle of which is shown in Fig. 31. It is still sometimes used in small engines. TS moves about T, PQ about Q, and PSH is a rocking

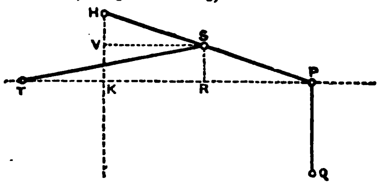


FIG. 31.

beam, of which the point H moves in a straight line HK. To find the relative proportions of the parts TS, PS, and HS, assuming that the arc described by P is almost a straight line,

Let the angle STP be θ; SPT, ϕ; then TK = TS cos θ − HS cos ϕ. Now, if H is to move along HK; TK = TS − HS; hence,

$TS - HS = TS \cos\theta - HS \cos\phi \therefore TS\,(1 - \cos\theta) = HS\,(1 - \cos\phi)$—that is

$$\frac{TS}{HS} = \frac{\sin^2\phi}{\sin^2\theta} \quad . \; . \; . \; (1)$$

Also

$$\frac{TS}{SP} = \frac{\sin\phi}{\sin\theta} \quad . \; . \; . \; (2)$$

Equations which enable us to eliminate θ and ϕ when these are very small. Thus

$$\frac{TS}{HS} = \frac{\phi^2}{\theta^2}$$

$$\frac{TS}{SP} = \frac{\phi}{\theta}$$

Hence

$$\frac{TS}{HS} = \frac{TS^2}{SP^2} \text{ or } TS.\ HS = SP^2.$$

We find that S P is a mean proportional between T S and H S.

If T S = H S = S P, and if P is compelled by guides to slide along the straight line T P for a short distance each stroke, the point H will move along the straight line H K, and the parallel motion is perfect.

110. Parallel motions require such good workmanship to prevent strains in the cylinder stuffing-box, that they are often dispensed with.

The end of the piston-rod may be attached to a block moving in guides, this block being attached to the end of the beam by a connecting-rod, or, when space permits, being connected with the crank directly.

In all cases of direct connection with the crank by means of a rod of ordinary length, the side-pressure on the slide is very considerable, and gives rise to much friction and great strains.

111. **Beam.**—When of cast iron, as it generally is, *the beam* is usually made in one casting. It is often built of wrought iron $\frac{5}{8}''$ plates, like an ordinary box-girder, the width of the box being about one-fifth of the depth.

Since the upper and lower flanges are subjected to nearly equal stresses in successive strokes, they are made equally strong. It may, however, be noticed that the weight of the beam counteracts the upward forces at F and K in the upward stroke, but it adds to the stresses of the downward stroke in straining the beam.

The length of the beam in stationary-engines is usually six times the length of the crank, its breadth at the middle being equal to the diameter of the cylinder, and its breadth at the ends being one-third of this.

CHAPTER X.

CONNECTING-ROD, CRANK, FLY-WHEEL.

112. BY means of the connecting-rod, the reciprocating straight-line motion of the piston-rod is converted, either directly or indirectly, into the circular motion of

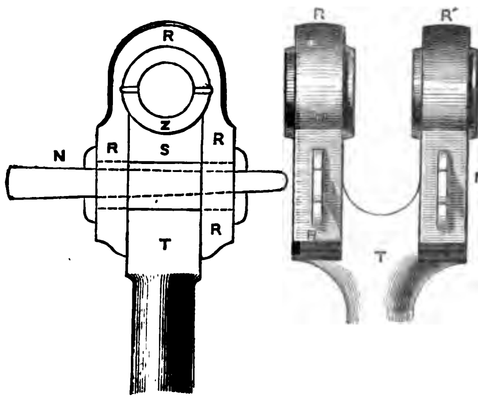


FIG. 32.

the crank. In the ordinary stationary-engine it clasps the beam by means of two jaws (Fig. 32), the brasses Z working round the *end-stud* which projects on each side of the beam. Consider one jaw: R is a strap, which by means of the cutter or key N and two gibs binds the brasses Z to each other, and to the block of the con-

necting-rod T. As in all other bearings, the brasses Z are kept separate by two thin plates of wood or metal, which may be made thinner as the brasses wear; so that, on striking the key, the brasses may again be made to touch the pin all round.

The strap, or some substitute for tightening the brasses, appears in the jaws of all connecting-rods, whether these jaws are single or double. In Fig. 33 may be seen a connecting-rod for locomotive-engines, in which, at the large end, a bolt is used, as well as a

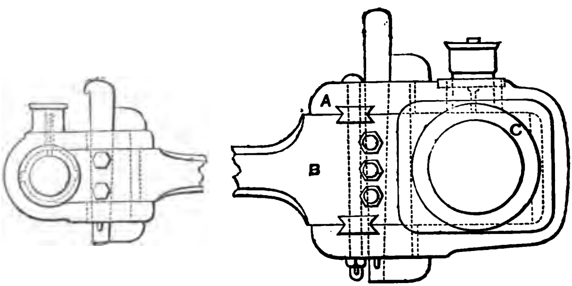


FIG. 33.

gib, to hold the strap on the block, three pinching screws and a little cutter being employed to prevent the great key, or cutter, from moving; the strap is flattened at the head for superior firmness; an oil-cup is shown in its proper position.

In Fig. 34, a simple connecting-rod end may be seen much used in the several pieces of a parallel motion. No strap being employed, the brasses have no flanges; they fit an octagonal seat, and are kept in their places by one key M, which passes partly through the brass and partly through the block. The brass N′ has a

projection fitting into the block, which prevents its sliding out laterally.

It is often found convenient to dispense with keys, and to depend altogether on bolts, as in Fig. 35, where the block and a cap are bolted together, holding the

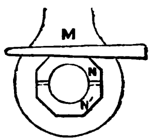


FIG. 34.

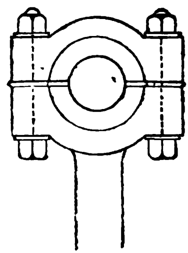


FIG. 35.

brasses between them, as in an ordinary stationary bearing.

Fig. 36 shows the head of a coupling-rod for locomotive-engines. Here the strap is really part of the block, or rather, there is no strap, the brasses being held in the ordinary way by a key. Two pinching-screws prevent the key from shifting.

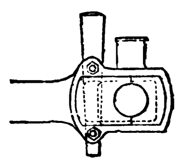


FIG. 36.

113. When it is not possible to gear the fly-wheel, so that a long crank-shaft is necessary, the cranks are double, like that shown at Fig. 37. B is the discontinuous shaft; C is the crank-pin, which is caught by the connecting-rod, and which has a circular path round the axis of B.*

* The engraver has made the shaft too large, hence the crank is weak-looking.

M and N are bored very accurately—a little smaller than the pin and shaft—and are fitted on when hot, so that on contracting they hold tightly.

In large mill-engines the fly-wheel has cogs, so that the shaft terminates with the crank, or cranks (two engines are now almost always coupled together); hence, single cranks may be used. The pin is overhung in single cranks; and in some horizontal engines, when the air-pump is worked from the crank-pin, this is overhung to a much greater extent than usual.

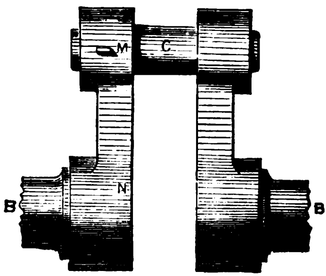


FIG. 37.

In inside cylinder locomotive-engines, and in many marine-engines, the double cranks are forged in one piece with the shafts, as in the crank of Fig. 38.

The work done by the steam in moving the piston is transmitted to the crank-shaft.

The connecting-rod transmits forces to the crank-pin which may be resolved in two directions, along and at right angles to the crank. Now, it is evident that during any instant the crank-pin is not moved in the direction of the radius; and hence the radial force is balanced, and does no work, all the work being transmitted by the tangential force.

We see, then, that so long as the pedestals of the crank-shaft are rigid, no energy is wasted through the medium of the radial force.

114. When the connecting-rod converts the motion of the end of the beam into the circular motion of the crank-pin, or when it directly converts the motion of the piston-rod into the circular motion of the crank, the relative velocities of its ends may be calculated at all times during the stroke. Since, in any train of mechanism, the velocities of two points are to one another as the forces exerted at those points in the directions of motion, and as the resistance to be overcome by the engine is practically constant during one stroke, these velocities will enable us to find the periodical excess and deficiency of energy given up of the fly-wheel during the stroke.

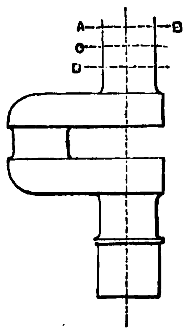


FIG. 38.

To compare the velocity of the point M (Fig. 29) with that of B—that is, to find the relative velocities of the piston and of the end of the beam.

Let the greatest angle which A B can make with its middle position be θ^1. At any other time let this angle be θ.

Let the vertical distance from M to the horizontal line from A at any time be called y, then it may be shown that

$$y = AB\sin\theta + \sqrt{BM^2 - AB^2(1 - \cos\theta)^2}$$

and therefore

$$\frac{dy}{d\theta} = AB\left\{\cos\theta - \frac{AB\sin\theta\,(1 - \cos\theta)}{\sqrt{BM^2 - AB^2(1\cos\theta)^2}}\right\}$$

from which we get

$$\frac{\text{velocity of piston}}{\text{velocity of end of beam}} = \cos\theta - \frac{\text{A B}\sin\theta\,(1 - \cos\theta)}{\sqrt{\text{B M}^2 - \text{A B}^2\,(1 - \cos\theta)^2}} \quad (1)$$

To compare the velocities of the extremity of the beam and of the crank-pin, and thus, by the help of the formula just found, to compare the velocities of the piston and crank-pin at any time,

Let the connecting-rod, when at a dead point, make an angle t with the vertical. Let it at any other time make an angle $t + s$ with the vertical. Let ψ be the angle made by the crank at any time, with the vertical. Let l = length of the connecting-rod, a that of the half-beam, and r that of the crank; then the distance from the end of the beam, when highest, to the centre of the crank, is $l + r$: and hence, as t is very small—

$$l + r = l\cos(s + t) + r\cos\psi + a(\sin\theta_1 - \sin\theta)\ . \quad (2)$$

Again,

$$l\sin(s + t) + a(2\cos\theta - \cos\theta_1 - 1) = r\sin\psi\ . \quad (3)$$

Also, the rod makes with the path of the end of the beam at any time the angle $s + t - \theta$, and with the path of the crank-pin $90 - \psi - s - t$; and from the principles of mechanism we know that

$$\frac{\text{velocity of end of beam}}{\text{velocity of crank-pin}} = \frac{\cos(90 - \psi - s - t)}{\cos(s + t - \theta)}$$

Now

$$\frac{\text{velocity of piston}}{\text{velocity of end of beam}} \times \frac{\text{velocity of end of beam}}{\text{velocity of crank-pin}} = \frac{\text{velocity of piston}}{\text{velocity of crank-pin}}.$$

Let us call this ratio R, then from (1) we get

$$\text{R} = \frac{\sin(\psi+s+t)}{\cos(s+t-\theta)}\left\{\cos\theta - \frac{a\sin\theta(1-\cos\theta)}{\sqrt{\text{BM}^2 - a^2(1-\cos^2\theta)}}\right\} \quad (4)$$

To obtain the ratio R in terms of ψ from these equations, θ and $(s+t)$ must be eliminated. To solve the question generally, the elimination will be rather long; but in special cases it becomes simple.

Special case.—Suppose that the beam is infinitely long, so that the relative velocities are as if the rod directly connected the top of the piston-rod with the crank-pin.

The reader may be reminded that

l = length of connecting-rod.
r = length of crank.
$\psi + t$ = angle turned through by the crank from its dead point.
R = ratio of velocity of piston to velocity of crank-pin.
f = distance passed through by the piston.

Then θ and θ' vanish, and $a = \infty$
In the limit $a(\sin\theta' - \sin\theta) = f$

The right-hand side of—

(1) becomes 1.
(2) " $l + r = l\cos(s+t) + r\cos\psi + f$ . . . 2*
(3) " $l\sin(s+t) = r\sin\psi$ 3*

Eliminating $(s+t)$ we get—

$$\cos\psi = \frac{(l+r-f)^2 + r^2 - l^2}{2r(l+r-f)} \quad . \; . \; . \; (5)$$

(4) becomes

$$\text{R} = \sin\psi - \frac{r\cos\psi\sin\psi}{\sqrt{l^2 - r^2\sin^2\psi}} \quad (4)^*$$

From (5) we can always determine ψ when given f, or f when given ψ. *From* (4)* *we can always determine* R, or

$$\frac{\text{velocity of piston}}{\text{velocity of crank-pin}}$$

when given ψ the angle made by the crank with the vertical. In Fig. 38*a*, $t = \text{O}\psi = \text{POB}$.

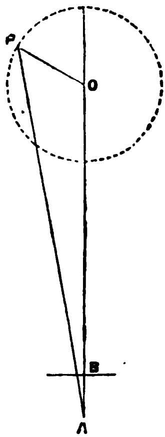


FIG. 38*a*.

Suppose l to become infinitely long, then from (4)*

$$\text{R} = \sin\psi$$

that is, that the ratio of the velocities of the piston and crank-pin is the sine of the angle turned through by the crank from the vertical.

Again, from (5), in the limit $\cos\psi = 1 - \frac{f}{r}$ that is, $f = r\,(1 - \cos\psi)$, which corresponds with our assumption in Art. 89 when discussing the slide-valve.

116. The crank-shaft works in two bearings when the fly-wheel is geared, otherwise it has three, one of which is placed near the spur or bevil-wheel, which transmits the motion. The pillow-blocks of these bearings, like the cylinders and pillars of the engine, rest on great columns of stone, which come up from the foundations, and are well bound by bolts to cast-iron foundation plates.

The fly-wheel has a heavy rim, the arms being sufficiently strong to prevent the bursting of the rim by centrifugal force. It is seldom that a large fly-wheel is cast in one piece, for adjacent parts vary so much in

thickness, that there is much inequality in the rate of cooling; and this would give rise to many unequal contractions, and consequently to permanent stresses: besides, the casting in many cases would be unmanageably large. Hence, large wheels, whether geared or not, are built in many pieces, which are fitted very truly to each other with keys and bolts.

As a rule, fly-wheels are geared, working into smaller pinions with mortised cogs.*

115. If P is the pressure on the crank-pin, at right angles to the crank, and R is the resistance at the same point to motion of the machinery, and if S is the arc travelled over by the pin, then $\int (P-R)ds$ means the excess of work performed by the steam; and although in a whole revolution this is equal to 0, it is not so when other periods of time are taken. The greatest value of this we shall call E, *the fluctuation of energy* given by the steam to the crank-pin. General Morin found E by experiment, and we give part of his table:—

Let

$$\frac{\text{length of connecting-rod}}{\text{length of crank}} = a$$

and

$$\frac{\text{fluctuation of energy}}{\text{whole energy given to crank in one revolution}} = b$$

* Mortised cogs are cogs of wood, generally of beech in the larger sorts of wheels, mortised into the iron rim, and held there firmly by two shoulders and a pin. Wherever a large wheel works a small pinion, and both revolve rapidly, it is thought well to use mortise teeth: first, because they prevent noise; secondly, their use saves trouble when the teeth of the wheel are broken—an accident which could otherwise only be remedied by getting a new wheel; thirdly, they give a superior smoothness of motion in the pinion.

Non-expansive.		Expansive Condensing when $a = 5$.		Expansive Non-condensing when $a = 5$.	
a	b	r	b	r	b
8	·105	3	·163	2	·160
6	·118	4	·173	3	·186
5	·125	5	·178	4	·209
4	·132	6	·184	5	·232
		7	·189		
		8	·191		

steam being cut off at $\frac{1}{r}$th of the stroke.

For double-cylinder engines, the ratios are almost equal to those of single-cylinder engines, working without expansion.

To find the fluctuation of energy during one revolution of the crank by means of these tables, *multiply the ratio there given by the whole work done by the steam during one revolution of the crank.* In future we shall consider E known, since we have shown how it may be determined.

118. Now, if v_1, v_2, and v_0 are the greatest, least, and mean velocities per second of the crank-pin,

$$\frac{v_1^2 - v_2^2}{2g}\int n^2 \text{W} = \text{E}.$$

Here W = weight of a small part of the engine which is revolving, n

$$= \frac{\text{radius of circle in which the small body moves}}{\text{length of crank}},$$

and $\int n^2$ W means the sum of all such terms as $n^2 \times$ W.

Now, the *mean energy* E_0, stored up in the revolving bodies $= \frac{v_0^2}{2g}\int n^2 w$, and $\frac{E}{2E_0}$ is found to be $\frac{v_1 - v_2}{v_0}$, which is the ratio of the fluctuation of velocity to the mean velocity, and hence may be called the *co-efficient of unsteadiness:*

That is, $\frac{1}{m}$ the co-efficient of unsteadiness $= \frac{gE}{v_0^2 \int n^2 w}$.

As a rule, the co-efficient of unsteadiness $\frac{1}{m}$ will vary from $\frac{1}{30}$ for ordinary machinery to $\frac{1}{60}$ for machinery for fine purposes.

Now, in fly-wheels, $\int n^2 w = \frac{R_1^2 W}{R^2}$, where R_1 is the mean radius of the rim of the wheel, W the weight of the rim, and R the length of the crank. Hence the co-efficient of unsteadiness $\frac{1}{m} = \frac{gE}{v_0^2 \cdot \frac{R_1^2 W}{R^2}}$. As $\frac{v^0}{R}$ represents the mean angular velocity per second in circular measure, we get from this *a rule for determining the rim of a fly-wheel:*—

$$R_1^2 W = \frac{mgE}{a^2}.$$

Where R_1 is the mean radius of the rim of the fly-wheel, W the weight of the rim in pounds, where m is a number varying from 30 for ordinary machinery to 50 for machinery for fine purposes; $g = 32{\cdot}2$, a is the mean angular velocity of the wheel, or the number of revolutions per second divided by 6·2832, and E is obtained from the tables already given.

It will be remembered that the relative values of R and W in a fly-wheel ought to depend on the fact that the centrifugal force should be as small as possible. Now, centrifugal force for a certain angular velocity varies as W R_1, and from the equation just given, $R_1 W = \frac{m g E}{a^2} \div R_1$ and that this may be small, R_1 ought to be great. *Hence it will always be found convenient to make the radius of the fly-wheel as great, and the weight of the rim as small, as possible.*

Although foreign to our subject, it ought, perhaps, to be mentioned here that for punching and slotting machines, the fluctuation of energy during one operation is almost equal to the whole energy involved in the operation; so that E in the above formula will in such a case represent the whole energy of one operation.

CHAPTER XI.

THE GOVERNORS.

116. THE governors of ordinary stationary engines are usually made on the principle of that shown in the general drawing, and in Fig. 39. A strong support, to which the balls and arms B, B C, and M N are attached, is whirled round by means of a band and pulleys, or other gearing, worked from the crank-shaft.

The arms M N are attached to a piece N T R, which slides up and down the support, and carries with it the end of the bell-crank lever T V.

At a certain speed of the engine the balls have sufficient centrifugal force to keep the sliding piece in a certain position, so that the throttle-valve, or other

regulator, which it works, admits a certain sufficient quantity of steam. When the mean pressure on the piston during the stroke is greater than the resistance to be overcome—that is, when the engine is doing less

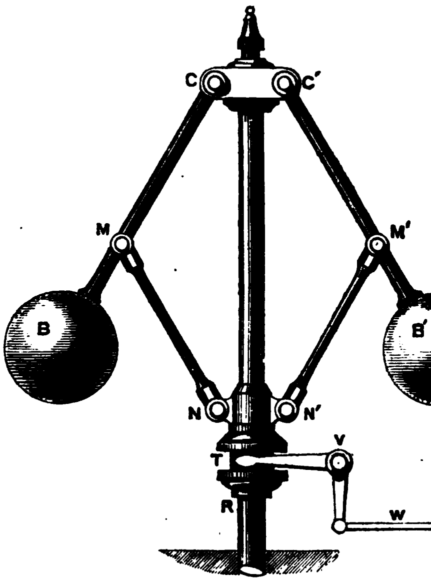


FIG. 39.

work than before—the speed of the engine increases, giving increased centrifugal force to the balls B, which lift the piece T, and so reduce the mean pressure of steam.

The support is more usually hollow, and a long rod passes down through the top, its lower termination being opposite a slot, through which it is connected with the arms outside, without itself being made to turn. Fig. 14 will explain the general nature of its connections with the throttle-valve (Art. 79), or other regulated means of admitting steam. *e d* and *b c* are two bell-cranks, or arms, at right angles, firmly keyed to horizontal shafts; and these, with the rods P, *d c*, and *a*, complete the regulating mechanism. (*See* Art. on Tappet Motion.)

There is a neat arrangement sometimes employed, consisting of a series of small collars at the bottom of the revolving spindle. The spindle when lifted moves a sector on the valve-shaft to which it acts as rack.

117. It may easily be shown from a principle of Mechanics (the triangle of forces), that if r is the radius of the circle swept through by the balls, h the vertical height from the balls to the point where the arms M C meet the axis, then

$$\frac{h}{r} = \frac{\text{weight of balls} + \text{resistance of valve, \&c.}}{\text{centrifugal force.}} \quad . \; . \; . \; (1)$$

Now, if there is no friction in working the valves, and if N is the number of revolutions of the governors per second, from the equations just given we get

$$h = \frac{9{\cdot}7848}{N^2} \quad . \; . \; . \; (2)$$

118. Thus *the height varies inversely as the square of the number of revolutions, so that if the speed gets greater, the height becomes less; and for any given height there is a definite speed.*

Now, it is necessary with a perfect governor to get the same speed with very different amounts of change in position of the regulating valve; but in the common

governor different amounts of such changes are only given by changes in the height of the balls. Hence, with ordinary governors, it is impossible accurately to get the engine to work at the same speed with different amounts of change in the regulating valve—that is, with different amounts of admission of steam and with different loads on the engine. This defect may be greatly remedied by allowing very small changes in the height of the balls to produce very great changes in the admission of steam. Another important remedy, more satisfactory even than the last, is the attachment of a heavy weight to the rod which is lifted, or to the piece N T R (Fig. 39). From formula (1) it may be seen that by the latter expedient we considerably increase h for a given speed of the governor; hence greater differences in the position of the balls are now consequent on ordinary changes in the speed. This is *Porter's Loaded Governor*, whose balls are usually small, and revolve at the rate of 200 turns per minute. The height may be calculated from Formula (2), which now assumes the form

$$h = \frac{9 \cdot 7848}{N^2} \cdot \frac{W + w}{w} \quad . \; . \; . \; (3)$$

Where

h = vertical distance from the centres of the balls to the point at which the arms B C meet the axis.
N = number of revolutions *per second.*
w = weight in pounds of the two balls.
W = sliding weight + friction of the throttle valve, &c.

The friction may in general be neglected.

These governors are much used. They are still imperfect, for they are not isochronous—that is, they do not maintain the same speed for all admissions

of steam, or for all loads on the engine. Now in theory it is necessary to alter the admission of steam without altering *h*. If the points of support can be made to move upwards as the balls move, it is possible to construct a perfect governor. This is effected in the Parabolic Governor, where the end of the rod M C is made elastic. In Fig 40, C is, as before, the point of support of the arm C B, which is flexible from C to F, in order that it may lie along the curved piece C L. Now, C L is the evolute of a parabola; hence the curve K B must be a parabola; and from the nature of this curve the vertical height from B to the point where B C cuts the axis must be the same in all positions of B. In constructing this governor, a speed is chosen; the sliding-weight and the weights of the balls are assumed. From these *h* may be calculated according to Formula (3) given above. A parabola is drawn such that the distance between its focus and its vertex is $\frac{1}{2}$ *h*. The evolute will be touched by every normal to the curve, and this may give a useful method of drawing the evolute. Any normal to the curve may be chosen as the medium position of the arm B C for ordinary loads.

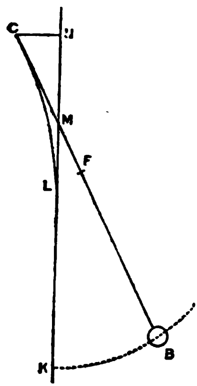


FIG. 40.

In Farcot's Governor, a very good approximation to the perfection of the parabolic is obtained by suspending C B from a strong arm N C.

Other isochronous regulators than the parabolic have been constructed. Siemens uses an epicyclic train of

three wheels to drive the governor. Any tendency to change the position of the ball, by increasing the speed of the engine, is resisted by excessive friction at the point of attachment. The sluggishness of the governor is immediately taken advantage of by the middle wheel, which is usually prevented from running round the other two by a balance-weight, but which now changes its position at once, and so affects the admission of steam.

119. Revolving fans are sometimes employed, to whose motion, at a certain speed, the air offers a certain resistance. When the speed is slightly increased, this resistance also gets greater, and the increased resistance is employed by means of springs, or otherwise, to affect the steam-valves.

In the Allen Governor, designed by Mr. Hunt, of Boston, a small paddle-wheel rotates in a box of oil, the box having projecting ribs to resist motion of the fluid; and hence turning with the wheel when the speed is great enough. When the box turns, it lifts a weight and closes the steam-valve. This governor is isochronous. In some experiments very considerable additions were made to the load on the engine without perceptible effect on the speed.

It is always driven by gearing. The weight is slightly altered by means of springs when lifted above or lowered beneath the mean position.

120. In marine-engine governors all weights are balanced, or else springs are substituted. When in the old governor the arms M C are produced beyond C, and provided with balls, which swing in a circle equal to that of B, we get an isochronous governor when the springs, used instead of a sliding weight, offer resistances which vary with the radius of the circle in which the balls swing.

CHAPTER XII.

BOILERS. DESCRIPTION.

121. **The Waggon Boiler**—now obsolete—was heated by means of a furnace placed beneath it, and of flues, allowing the heated gases to pass round the sides merely, or else through the middle of the boiler, and then round the sides. The low pressure of the steam enabled a self-acting contrivance for feeding the boiler to be employed.

The Cylindric Egg-ended Boiler is heated in the same manner. It is found that sediment is very apt to collect on the bottom plates, just above the furnace, so that these plates become non-conducting in time, and are liable to burn.

The French Boiler has two long wide water-pipes, extending through the furnace and bottom flue, connected with the bottom of the boiler by means of a number of smaller pipes. After passing through the bottom flue, along which the cylindric water-heater extends, the heated gases are conducted to the front by the inner of three flues placed under the boiler proper; splitting in front, and passing to the chimney by the two side flues. This boiler has many advantages, and, when provided with air or steam jets passed through the bridge, is very efficient.

Thierry exhibited, in 1867, a good form of this boiler, provided with steam jets. Lecherf, at the same time, exhibited a boiler in which there was only one flue, placed beneath, the return being made by 26 internal tubes, and the gases being led to the chimney by the ordinary side flues.

122. **The Cornish and Lancashire Boiler** (Figs. 41 and 42).—This is the most common form of boiler employed for stationary engines. The *shell* is cylindrical, with flat ends, and one or two cylindric *flues* pass from end to end.

The diameters of these flues, when single, are $\frac{2}{5}$ths of

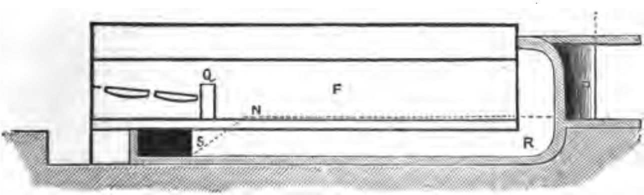


FIG. 41.

the diameter of the boiler; when double, $\frac{3}{8}$ths of the diameter of the boiler. The heated gases pass over the bridge Q, on their way along F, and downwards at R to the flue beneath the boiler seen at R S and K. At S they divide into two parts, each going along a side flue to join the other at P, before it reaches the chimney. The *damper* is placed at P, to enable the engineer to regulate the draught; it enters from above, being suspended by a chain, which also passes over a pulley, and carries a balance-weight.

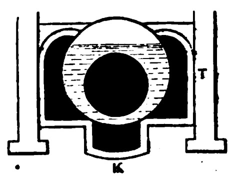


FIG. 42.

The *bridge* is generally of fire-brick, resting on the bottom side of the flue. It produces beneficial eddies in the stream of gases, confines the coals to the furnace, and enables the furnace to act as a chamber in

which the oxygen and combustible gases may be conveniently mixed.

Sometimes, in other boilers than this, the bridge is of iron, and contains water. When the bridge comes down from above, it is said to "hang;" when there is a space above and below, it is called a "mid-feather."

The flue behind the bridge is sometimes called the "flame-chamber," where it is necessary to maintain a high temperature until the gases have got thoroughly mixed together. This chamber may, with good results, be lined with fire-brick, and useful eddies may be produced by proper arrangements of fire.

It will be shown presently that it is well to produce eddies in the heated gases by obstructions in the flues, and by sudden turns, as these enable the gases to give off their heat readily, and as they help to mix oxygen with the combustible gases before they get too cool for ignition. The downward flue at the end of the boiler R creates a very useful eddy, and water-tubes crossing in different directions, or passing along the flue, greatly add to the effective heating-surface.

With two cylindric flues there is an increase of efficiency, for the firing of the two furnaces may be alternate. The advantages arising from this are discussed in Art. 142. Why these double flues should be separate for the whole length of the boiler in many modern examples it is difficult to see, when so many advantages would arise from their union behind the fire-bridges. Certainly, there is no difficulty of construction in the way. In the best flue-boilers this union is made, and the Cornish boiler was perfected when the gases from the two furnaces were mixed in a chamber behind the bridges, and were then made to pass through a number of tubes. This is the most efficient form of furnace for stationary engines.

Whenever tubes are employed in boilers, they are

supported at the ends by tube-plates, into which they fit.

To obtain good firing, and to prevent the weakening effect of a large flue, the furnace part is often greatly strengthened and made larger, so that one or more tapering plates are introduced.

The Messrs. Galloway exhibited, in 1867, boilers with very efficient furnaces, in which many water-tubes were introduced into an elliptical flue (the greater diameter of which was horizontal), the furnace parts being of the ordinary cylindric shape.

Thomson's Land Boiler is provided with tubes, and in many ways its construction resembles that of a marine boiler.

Many designs have been made for compound boilers, consisting of a number of separate strong tubes, communicating at the bottom and top. One of the best of these is that of Howard, exhibited at the Paris Exhibition.

In a tube-boiler at the Vienna Exhibition there are twenty nine-inch tubes, 9 feet long, of welded wrought iron, arranged in five horizontal and four vertical rows, lower in front than rear. The rear ends of each vertical row are connected by a vertical tube, and the lowest row in front by a horizontal feed-water pipe. The steam-chest communicates with the four vertical tubes by means of small pipes. The furnace is beneath, the flame surrounding the tubes.

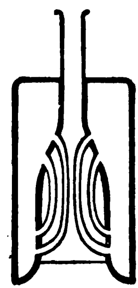


FIG. 43.

123. **Vertical Boiler** (Fig. 43.)—This usually consists of a cylindric vessel, with its axis vertical, the furnace-bars being placed near the bottom, the furnace being axial, and surrounded by water on all sides.

The furnace gradually narrows into the chimney at the upper part, and through it pass many bent water-tubes, connecting the upper and lower parts of the boiler.

Napier's vertical boiler has a cylindrical shell, with a hemispherical top, the large inside flue—in which there are many vertical and some horizontal water-pipes—also being cylindrical.

Meyn's vertical boiler at the Vienna Exhibition has a fire-tube surface besides the water-tube and other heating surfaces mentioned above. The heated gases pass through the water and steam to the chimney by small

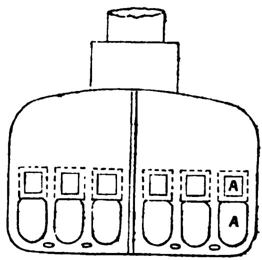


FIG. 44.

tubes. The water-tubes are corrugated, and have an oval section.

124. **Locomotive Boilers.**—These will be more particularly described in Book III. Fig. 55 shows the construction generally adopted.

N is the fire-box, O the barrel, and G the smoke-box. Fuel, which is coke or coal, or a mixture of both, is laid on the fire-box X, through the double-door $P^1 Q^1$. The fire-box is surrounded on nearly all sides with water-spaces. The hot gases go along the tubes O, of which there are a great many, giving off their heat

in passing through the barrel of the boiler on their way to G, the smoke-box, from which they escape by the chimney P. The draught is helped, or almost wholly created, by the escape of exhaust steam into the chimney from the blast-pipe H.

128. **Marine Boilers.**—These are either flue or tubular boilers. Figs. 44 and 45 give an idea of the general configuration of a flue-boiler. It will be seen that as many eddies as possible are produced in a

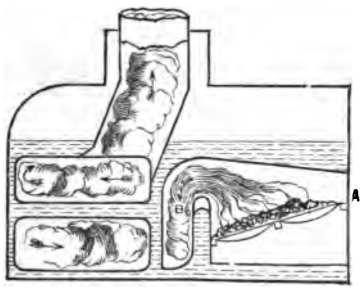


FIG. 45.

flue-boiler of such limited size. This construction is, perhaps, too cumbrous for steam-vessels, yet many people prefer it to the other. In the Cunard line of steam-ships it is well thought of.

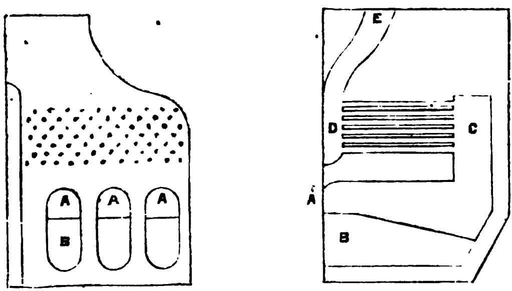
Figs. 46 and 47 give an idea of the general configuration of a *tubular* boiler. In front of the tubes at D there is a door, which may be opened to get the tubes cleaned.

Napier's boiler, designed for certain small steamers on the Clyde, has small vertical water-tubes in the flues

and resembles some vertical land boilers. It is very efficient.

In some very efficient boilers, constructed by Rowan, of Glasgow, the flues are vertical, and consist of thin rectangular spaces, well stayed by means of rectangular cross-pipes.

129. When mineral oil is used as fuel, it is injected in a very divided condition into a fire-brick chamber,



FIGS. 46 and 47.

together with a sufficient supply of air for complete combustion. Sometimes the oil is allowed to enter by the force of gravity, the air entering otherwise. Sometimes a jet of superheated steam forces the oil in a fine spray along with the air into the chamber.

These expedients are not likely to be of great ultimate importance, as the supply of mineral oil is very limited.

CHAPTER XIII.

SUPERHEATING, DRAUGHT, AND GRATE.

130. **Superheating.**—It will be shown in Art. 163-180, that engines employing superheated steam give more efficiency than when the same engines employ the same steam not superheated. In fact, this may be seen from the principles of Book I., Art. 41; for it is there shown that, if an engine gets a quantity of heat H, from the source, the work producible by a perfect engine is

$$\frac{J\,H\,(T - T^1)}{T}$$

where T is the absolute temperature of the fluid as it leaves the source of heat and T^1 that of the refrigerator, and where J is Joule's equivalent.

Hence the available amount of work to be obtained from the fluid depends on the ratio

$$\frac{T - T^1}{T}$$

Now, T is already very great, compared with $T - T^1$. Hence, by increasing T, we increase the ratio—that is, the maximum amount of work obtainable from the heat H. *Hence, when the temperature of the source, and therefore that of the working fluid, is increased, the efficiency of the engine is increased.*

For example, let steam be at the temperature of 127° C., let the condenser have a temperature of 40°.

Then

$$\frac{T - T^1}{T} = \frac{87}{273 + 127} = \frac{87}{400} = \cdot 2175.$$

Now, let the steam have a temperature of 137°, that of the condenser remaining as before. Then

$$\frac{T - T^1}{T} \text{ becomes } \frac{97}{410} = \cdot 236,$$

a ratio greater than the last. On giving the same quantity of heat to steam in these two cases, the maximum quantity of work which can possibly be obtained is greater in the second than in the first case by the ratio

$$\cdot 236 : \cdot 2175.$$

Wetherhed, of America, first introduced superheating. He passed the steam through two pipes, superheating that in one passage, and afterwards letting it mix with the other. It is now found more convenient to raise the temperature of the whole mass at once, and never to exceed 170° C., as above this there is liability to spoil the piston and other packings, and the valves. The steam is passed through a chest of small tubes, placed in the chimney, or in the up-take. Logs of vessels, before and after the introduction of superheating, give ample evidence of the superior economy effected by its means.

In the boilers of the *Great Eastern*, Watt passed the steam round small pipes, through which the smoke passed; other arrangements have also been adopted, but the method just described, which is that adopted by Napier, is the one still generally employed.

131. **Draught.**—It was shown (Art. 52) that the volume of heated products of combustion is very

nearly equal to the volume of air supplied, when there is the common inconsiderable amount of hydrogen in the fuel. Now, there may be 12, 18, or 24 lbs. of air supplied per pound of fuel (*see* Art. 52); and these weights correspond at 10° C. to 150, 225, and 300 cubic feet respectively.

Given the weight of fuel burnt every second to find the necessary draught-pressure, or the urging required for supplying the furnace with air.

If w is the weight of fuel consumed *every second*, if k is the volume of air supplied per pound of fuel (as shown above—it usually varies from 150 to 300 cubic feet), the velocity of the current in the chimney in feet per second

$$v = \frac{w\,k}{\mathrm{A}} \cdot \frac{t + 273}{283} \quad . \; . \; . \; (1)$$

where t is the temperature of the gas in the chimney, and A the sectional area of the chimney or flue.

A pressure producing motion in a fluid is conveniently expressed as the height or "head" of the fluid in question which is necessary to produce the pressure.

Peclet's experiments show that if h is the "head" of heated air required to produce the above velocity v, then

$$h = \frac{v^2}{2g}\left(13 + \frac{0{\cdot}012\,l}{m}\right) \quad . \; . \; . \; (2)$$

where $g = 32{\cdot}2$ feet, $l =$ the length of the flue and chimney

$$m = \frac{\text{area of cross-section of the flue or chimney}}{\text{the perimeter of cross-section of the flue or chimney}}$$

m, in fact, is what is usually called the hydraulic mean depth.

When w and t are given, v may be calculated from (1), and when v is known, and the dimensions of the

flue, h may be calculated from (2). Knowing h, it is easy to calculate the pressure in some other form. In fact, it may be shown that *if* P *is the pressure in pounds per square foot*, producing the draught in question,

$$p = \frac{283\,h}{273 + t}\left(\cdot 0807 + \frac{1}{k}\right) \ldots (3)$$

The draught may be produced by fans or blast-pipes, or by means of a chimney. In all cases where the efficiency of the blast-pipe or fan is known, the work done in maintaining the velocity of v (3) may be calculated. (*See* Book III. on the Blast-pipe of Locomotives.)

132. In *Chimneys*, let H = the height of the chimney. Let us suppose that the temperature of the external air is T° C.; then it may be shown that

$$H = h \div \left(\cdot 96 \times \frac{273 + t}{273 + T} - 1\right) \ldots (4)$$

From (4) we may calculate the height of chimney necessary to produce a given draught; that is, by means of (1), (2), and (4) questions on air supply and height of chimney are completely solved.

It may be shown from (4) *that the best chimney draught is obtained when*

$$\frac{273 + t}{273 + T} = \frac{25}{12}$$

t being the temperature in the chimney, and T that of the external air. When this condition is fulfilled, the "head" h is equal to the height of the chimney.

When T = 10° C., it may be shown that for greatest efficiency t = 317° C., that is, *the temperature of the hot gases in the chimney ought to be nearly equal to that of lead just melting.*

132*a*. *Draught required for a particular furnace.* The following table gives the amount of coal in pounds burnt per hour per square foot of fire-grate in different furnaces, according to the best practice :—

Character of Furnace.	lbs. of coal per square foot per hour.
Cornish boilers for pumping engines .	4 to 10
Factory boilers (Cornish and others) .	12 to 16
Marine boilers, ordinary rates . . .	16 to 24
Quick rates of ordinary factory and Marine boilers, with chimney draught	20 to 27
Locomotives	40 to 120

Knowing, then, the area of the particular furnace, w is easily calculated from the table. Employing equations (1), (2), and (3) given above, we may determine the draught as a pressure. Employing (1), (2), and (4), we may determine the height of chimney necessary. Indeed, in determining the height of the chimney in ordinary cases, we need only use (1) and (2), since $\text{H} = h$ when the temperature of the heated gases in the chimney is 317° C.

133. **Grates.**—It will be seen below that it is necessary for high efficiency that the quantity of fuel burnt per hour per square foot of fire-grate should be properly proportioned to the draught. This is often best discovered by practical trials with any given boiler, firebricks being introduced from time to time as the furnace is found to be too large. It is found in practice that a smaller grate is needed for a certain quantity of coal

when air enters above the fuel than when it comes up through the grate.

No grate in any boiler is ever much longer or shorter than 6 feet, the breadths varying from 15 inches to 4 feet. It consists of fire-bars, of two or three feet in length, and the cross-bars by which these are supported.

Wherever possible, furnaces should be high, as the crown of a low furnace is apt to get too cool from entering gases, and as the gases ought to get properly mixed.

CHAPTER XIV.

BOILER APPENDAGES.

134. **The Dome** is a cylindric vessel, with a spherical top, employed in locomotives to give additional steam-room to the boiler, and to be otherwise useful in preventing priming. There are also additions to the steam-spaces of marine boilers on the same principle.

Skilful engineers differ in opinion as to the proper amount of steam-room to be allowed.

With much steam-room we have freedom from priming and regularity of pressure, which would otherwise vary at every stroke of the engine. With small boilers, however, the quantity of water which may be heated at the same time ought not to be reduced too much. The steam-room is made to vary from one-fourth to one-half of the whole capacity of the boiler; a cylindric flue-boiler being usually filled with water to about three-fourths of its depth.

The Man-hole is large enough to allow a man to enter the boiler. The frame is riveted to the shell by

flanges, its well-fitting cover being bolted on. This cover is sometimes inside, being kept in its place to a great extent by the pressure of steam. **Mud-holes** are placed at the lowest parts of the boiler, that it may be cleaned out when necessary.

The Blow-off Cock enables us to discharge muddy water and sediment; it is placed at the bottom of the boiler. In many marine boilers much use is made of blow-off cocks to discharge brine. Others near the surface of the water discharge scum.

135. **Feed-water** is usually supplied by the *feed-pump* when the engine is in motion, the *feed-pipes* being provided with a heavily-weighted valve which allows the water to escape to a cistern when the engineer closes the *feed-cock*. When the engine is at rest, the boiler is fed by water-pressure from a cistern, or by means of donkey-engines, or with an injector. (*See* Book III.)

The amount of feed necessary for stationary boilers is usually more than twice that obtained by us in calculations in Art. 172, because we made no allowances for priming, blowing-off, &c. In marine engines it is necessary to have pumps which will lift from three to four times the calculated amount.

The feed-water is often heated in a *water-heater*, consisting of sets of tubes placed in the path of the heated gases. The soot is removed by scrapers worked from a pulley and band.

136. **The Pressure Gauge.**—This is of many forms, and is usually attached to the front of the boiler, so that the steam-pressure may be observed at any time.

The Mercurial Gauges used about engines are of four varieties :—

(1.) A **U** tube, the lower part of which in both limbs is partially filled with mercury. One limb is connected

with the boiler, the other being open to the atmosphere. Difference of level in the tubes determines the pressure. A difference in height of 29·92 inches is equivalent to the pressure of one atmosphere, or 14·73 lb. per square inch.

(2.) A U tube similar to the last, but smaller, and closed to the atmosphere, so that pressure may be measured by the amount of compression of a certain quantity of dry air.

(3.) A long vertical vacuum-tube (Fig. 48) one end of which dips beneath the surface of mercury in a basin, the other being connected with the condenser, or other vessel, the pressure in which is required to be known.

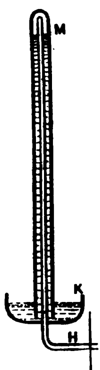


FIG. 48.

(4.) A vacuum gauge, consisting of an ordinary barometer tube, the surface of the liquid in the basin being placed in connection with the fluid, the pressure of which is required to be known.

It is evident that the scales of all mercury gauges ought to be exactly vertical. As mercury gets less heavy by heat, the height of the column must be corrected for temperature; this correction will readily be made by those who know the principles of Book I.

Bourdon's Pressure Gauges are now almost universally employed for all purposes. They are made very exact, both for condensers and for high-pressure boilers.

A long spiral tube (Fig. 49), with an oval cross-section, is closed at one end, and at the other communicates with the vessel the pressure which is required. If the pressure is less than that of the atmosphere, the spiral gets a quicker curve. If the pressure is greater than that of the atmosphere, the spiral tends to be-

come straight. The motion of the end of the tube is communicated, by means of a series of levers, or wheels, to an index which is graduated by the manufacturer by means of very accurate mercury gauges.

A pressure gauge, giving continuous indications of

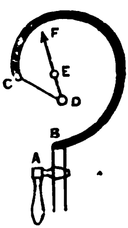


FIG. 49.

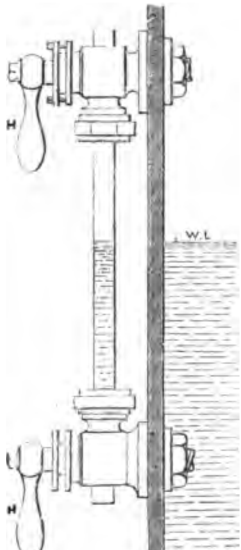


FIG. 50.

the steam-pressure in the boiler, has been designed by Mr. Edson, of New York.

137. **The Water Gauge.**—Three cocks placed at different heights in front of a boiler show, when turned,

whether the water is too low or too high. With these there is always used another gauge (Fig. 50). which consists of a glass tube, communicating at its ends with the steam and water respectively. It shows, by the level of the water visible within, the level of the water in the boiler.

In Germany there has recently been introduced a glass gauge which has two tubes, one inside the other. There is less likelihood of fracture, and the inner tube is protected from rapid changes in temperature.

138. **The Safety Valve.**—That the steam-pressure may never exceed a certain limit, and that this may be effectively provided for, there are two safety-valves

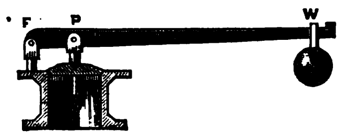


FIG. 51.

on the boiler, the common forms of which are shown in Figs. 51 and 52, or roughly at S and R (Fig. 56.)

Fig. 51 shows the form commonly adopted in stationary and in many marine boilers. The lever F W has F for its fulcrum (F ought, if possible, to be provided with a knife-edge), P for the point at which the steam-pressure on the valve is applied, and W the position of the weight. The pressure on the valve must overcome the weight of the valve, the weight of the lever, and the weight at W before steam will escape. Hence, if a is the area of the valve, and w'' its weight, p the pressure per square inch in the boiler, and w the weight at W; if w' is the weight of the lever,

and G F the distance of its centre of gravity from the fulcrum—

$$(p\,a - w'')\,\mathrm{P\,F} = w\,\mathrm{W\,F} + w'\,\mathrm{G\,F}\ \ldots\ (1)$$

from which all calculations regarding safety-valves may be made.

The lever is graduated, different distances from F representing different pressures at which the steam will escape. This graduation is most accurately made by means of special experiments.

138*a*. **To Graduate the Lever.**—From the formula given above—

$$\mathrm{W\,F} = \frac{p\,a.\,\mathrm{P\,F}}{w} - (w''\,\mathrm{P\,F} + w'\,\mathrm{G\,F})\ \ldots\ (2)$$

So that equal differences in the pressure p are represented by equal differences in the position of the weight.

Now, given a, w, w', w'', G F, and P F, find W F when p is, let us say, 20 lb. per square inch. Again, find the distance from F to the weight when p is 100 lb. per square inch, and divide the distance between the two positions into 80 equal parts each, to represent one pound of difference in the pressure per square inch.

In safety-valves provided with springs at W the distance W F never changes.

139. The weighted safety-valve is never employed in locomotives, and but seldom in marine-boilers, as much steam is liable to escape during the oscillatory motions of the machinery.

Spring-loaded valves are in many cases of the shape shown roughly in Fig. 56, in which the spring itself is hidden. One end of the spring is attached to a stud on the boiler, the other end being attached to the end of the safety-valve lever by means of a screwed rod held by a nut.

It was found that great weights and strong springs

were necessary in small valves, as great lengths of the lever permitted only a small escape of steam, often quite insufficient for the safety of the boiler when the pressure became very great. It is now usual to provide at least one valve of another kind: this being pressed upon directly by means of a spring, as shown in Fig. 52, or roughly in Fig. 56.

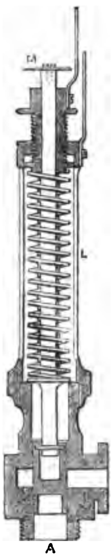


FIG. 52.

The scale on the case containing the spring is graduated by special experiments, a few points being found in this way, and others interpolated.

Nasmyth and Fairbairn load from the inside of the boiler.

A marine-boiler safety-valve, patented by Watson and Andrews, consists of a ball and seat (Art. 78), the ball being loaded from inside the boiler, and the seat having grooves to contain lubricants.

It is sufficient to make the safety-valve of such a size that it lets steam escape as fast as it is generated. This is accomplished when the area of the valve in square inches is from $\frac{1}{25}$th to $\frac{1}{30}$th of the weight in pounds of the coal burnt per hour, or is equal to $\frac{5}{8}$th square inch + the normal H.P. of the engine which it supplies with steam.

A *safety-valve* requires a higher pressure to keep it well open than that which lifted it. Ashcroft's valve, Turton's valve, and others, have been designed to remedy this defect.

Marine boilers are usually provided with a locked-up safety-valve, placed beyond the control of the engineer;

but it is found that, unless the valve is periodically lifted from its seat, it will stick fast; and hence the expedient is looked upon as misleading. In all cases one open safety-valve, at least, ought to be under the control of the engineer, and for this purpose a system of rods and levers passes from the valve to the firing-stage.

140. **The Vacuum Valve** is a safety-valve, opening inwards, for admitting air and preventing collapse in the shell when the pressure in the boiler is much less than that of the atmosphere. Diminution of pressure in the boiler may arise from condensation of steam by gradual cooling after the fires are withdrawn, or in marine boilers by the sudden shipping of water.

141. **The Fusible Plug** stops a hole in the fire-box, or crown of the furnace, and melts when the temperature of the steam, and consequently its pressure, becomes too high. An alloy of bismuth, lead, and tin melts at 98° C. Others of tin and bismuth, or of tin and lead, melt at temperatures varying from 119° to 168°, according to their compositions. The melting of the fusible plug cannot be well relied upon.

CHAPTER XV.

EFFICIENCY OF THE HEATING PARTS OF THE BOILER.

142. The efficiency may be defined as the relation which the quantity of heat given to the water bears to the total heat given out by the fuel. The causes of loss of heat may be classified in the following manner :—

I. *The loss from hot ashes and waste of unburnt fuel.*—Some coals are very brittle, and, to prevent waste of the small particles through the bars of the grate, the stoker should be directed to throw the coal on the fire in a uniform layer, which will need little stirring until it has caked; to clear the grate-bars always from below, and to sift the cinders for fine coal. It is sometimes found convenient to wet fine coal before throwing it on the fire. In Wales, the fine coals are sometimes rolled up into balls with water, or even mud. The loss from this cause, when there is careful firing, seldom exceeds 2½ per cent.

II. *The waste by unburnt gases and by smoke.*—In Art. 51 it was shown that an intimate mixture of the combustible gases of coal with the oxygen of the entering air was necessary for complete combustion. For coke and coals containing only a small weight of hydro-carbons, enough air will enter by the grate, and a proper mixture of the combustible carbon and carbonic oxide at the proper temperature of ignition will be made in the furnace if the fuel is evenly spread in moderate quantities at a time. When hydro-carbons form much of the weight of coal, a proper mixture of the gases is seldom obtained by the above method of firing. *Watt's* plan, which is still found to give the best results of all the simple methods, consisted in coking his coal in the front part of the furnace, so that the gases passed over the hot coke which filled the back part. This coke was burning in the ordinary way; that is, it was spread evenly on the fire-bars in a thin covering. Much more air was allowed to pass through the fire-bars than merely burnt the coke; this mixed with the gases at a high temperature above the incandescent coke just before passing the bridge, and hence the combustion was complete. The coked coal was soon spread evenly over the back part of the furnace, or rather more thickly towards the bridge,

and the front part was again charged. For the front of the furnace it is well to use a dead-plate when brittle coals are burnt.

Fairbairn used this principle in his double furnace, burning the gases of one grate freshly charged, above the incandescent coke of the other.

Williams admits air above as well as from below, by allowing holes in the fire-door to give a constant stream. This necessitates the constant charging of the furnace with fresh coal in small quantities at a time, and is quite successful when sufficient care is taken by the stoker. When, however, the furnace becomes filled with coke, as it may through negligence, the entering air has a great cooling effect, and, besides, is of no value as a supporter of combustion.

Juckes supplies coal with a chain-grate, which, by means of mechanism, gradually conveys it to the back as it cokes; and this, in connection with Williams' air-holes, proves a very efficient arrangement.

Boulton and *Watt* employed for the same purpose a circular revolving grate.

Maudslay uses round fire-bars, which revolve separately, and carry the coal to the back.

Bolzano's grate has two tiers of bars. Below the second tier is a flat plate, which may be pulled backwards. The furnace-door is a *hopper*, which spreads fresh coal evenly over the first tier of bars with a rake. All alternate bars may be lifted by means of levers, so that clinkers are gradually pushed towards the flat plate. When the plate is pulled back the clinkers fall into the ash-pit.

Williams ably advocates the entrance of air behind the fire-bridge as a means of completely burning coal; but this seems to have a bad cooling and heating effect on the boiler-plates.

Prideaux, by means of a self-acting, venetian-blind

fire-door, stops the stream of entering air as the charge of coal gets coked. He employs a slowly-descending piston, which, on the principle of the cataract (Art. 64), presses, slowly escaping air through a small orifice in the bottom of a cylinder.

Gorman closes the ash-pit, so that air cannot enter through the fire-bars. When a new charge of coals is in the furnace, he opens the holes in the fire-door; opening the ash-pit and closing the fire-door holes as the coking advances.

Clark injects air to the furnace at the level of the fuel by means of a number of little jets of steam, which enter through a cock placed within reach of the stoker. Where the draught is the same at all times, as in stationary boilers, the entering air is completely controlled by the limited use of Gorman's expedient; but where the draught is variable, as in locomotives, Clark's jets are very effective, and, indeed, may be considered the only proper expedient for locomotive engines.

A jet of steam from the boiler of a locomotive, admitted to the chimney when the speed of the engine is slackening, and when the blast is not working, may be usefully employed, according to Gorman's principle stated above. But it will be remembered that the absence of smoke with such a chimney does not prove that all the smoke is burnt: a great deal of it is too much saturated with condensed steam to go off in the usual manner. (*See* Book III.)

Dr. Marsh supplies *all* the air by jets above the firing-fuel.

The combustion-chamber has already been described in Art. 125. Here the air and gases mix at a proper temperature in a fire-brick chamber.

Combinations of the above methods are very often employed.

In the *Système Beaufumé* the carbon of the coal

is, in one fire-box, converted into carbonic oxide which is burnt by jets of air in a second chamber.

When insufficient air is supplied to furnaces, it seems that the greatest possible quantity of waste, even with bad firing, is the waste of the hydrocarbons; that is, instead of getting the heat due to one pound of coal, we get the heat due to the weight of coke in one pound of coal.

When insufficient air is supplied, but the firing is good—that is, in all the most ordinary cases of coal-burning in boiler furnaces—the heat obtained seems, from experiment, to be that due to the total quantity of carbon in the coal, whether it exists in the state of a hydrocarbon or not. Hence, instead of taking into account the hydrogen, which was in excess of that necessary to combine with the oxygen of the coal, as we did in Art. 53, we now suppose no hydrogen whatever to be burnt. The ratio of waste may be easily calculated.

Let there be fractions of a pound, c of carbon, h of hydrogen, and o of oxygen in 1 lb. of coal. Then, when there is no separate appliance for mixing the gases with air, this fraction—

$$\frac{4 \cdot 28\left(h - \frac{o}{8}\right)}{c + 4 \cdot 28\left(h - \frac{o}{8}\right)}$$

of the whole amount is wasted.

When there is also bad firing, and we may suppose all the hydrocarbons to be wasted, instead of 4·28 we must use numbers varying from 7·28 to 10·28, according as the hydrocarbons may be supposed to have the chemical formula $C\,H_4$, or $C_2\,H_4$. Hence, for coal of the average composition of that in Art. 52, the waste, when there is good firing but no separate contrivance for admitting air, is 18 per cent.

When there is bad firing in this furnace, the waste is from 27·7 to 35 per cent.

III. *External Conduction and Radiation.* — The radiation through the door when it is open, may be calculated by means of Art. 53, Book I., assuming, as we do, that the total heat radiated is equal to half the whole heat of combustion.

The conduction may be estimated when we know the temperature of the boiler at different places, the thickness and nature of the conducting body, and the temperature outside. In well-planned furnaces there is little loss of heat from conduction and radiation.

IV. *Heated gases escaping by the chimney.*—This is commonly the most important cause of loss of heat by the furnace. The hot gases usually escape at the temperature of 317° C., and the amount of heat carried off by the 300 or 600 cubic feet of gaseous products of combustion of 1 lb. of coal may be readily calculated.

When the chimney is of much greater cross-section than the area of grate, the temperature of the escaping gases is much less than 317° C. When the draught is produced by means of a blast-pipe or blowing-machine it may be shown that less energy is wasted than when produced by heated gases alone; hence greater economy is possible by using a blast, or Clark's jets,* into the furnace. But it is often advisable in stationary engines to have as high chimneys as possible, that the smoke and carbonic acid may be dissipated.

With a quick draught, less air is necessary for diluting the carbonic acid (which would otherwise prevent combustion), as the carbonic acid is now drawn away

* In a competitive trial of the steam-jet and fans for creating artificial draught by Mr. Isherwood (U.S.) on marine engine furnaces, the losses were, respectively, 9·73 and 1·41 per cent. of the whole expenditure of fuel. I do not know whether these experiments have been repeated.

more rapidly. It may be here noticed, not necessarily however, as a more economical condition, that the temperature of the furnace is higher in this case, and there is quicker giving up of heat to the water, and hence quicker evaporation.

In particular cases the question arises, What is the most convenient temperature to which the gases should be cooled? Irrespective of the necessity for chimney-draught, they ought, of course, to be cooled as much as possible; but an inconvenient length of flues may be necessary, or else the flues may be so much obstructed, that a great draught is necessary. It will be found that it is in general best to have the flow of gases much obstructed, as in locomotive engines, wherever it is possible to employ artificial draughts.

Experiments on boilers, made in 1872 by a Committee of the American Institute, show "that the introduction of boilers having exceptionally large proportions of heating-surface, and with large feed-water heaters, and depending upon a mechanical draught, will, when properly designed and constructed, be attended with a marked economy, which the Committee judge should more than compensate for the increased trouble and expense involved if large boiler-power is required."

The sectional area of the tubes or flues is usually made from one-fifth to one-seventh of the area of the grate.

From what has been said in Art. 49, it will be seen that we are now only concerned with the efficiency of the heating-surface, and from Formula (3)*, Art. 49, Rankine has determined that, when there is no appreciable loss from causes I., II., and III, nor from entrance of too much air, the efficiency of the furnace is—

$$\frac{S + aF}{bS}$$

Where a and b are constants, which may be determined by experiment.

S = No. of square feet of heating-surface per square foot of grate.*

F = No. of lb. of fuel burnt per hour per square foot of grate.

The values of *a* and *b* vary with the method of introducing feed-water, and of creating draught.

	b	*a*
1. Feed-water entering at the the cool part of the boiler, or heated in tubes in the uptake before entering, the draught being produced by a chimney	1	0·5
2. No special care taken of the feed-water, the draught being produced by a chimney as before	$\frac{11}{12}$	0·5
3. With best method of heating the feed-water, and with a forced draught	1	0·3
4. With no special care being taken of the feed-water, and with a forced draught .	$\frac{19}{20}$	0·3

The outside surface of the feed-water heaters must be included in S. It is to be understood, that although this rule is useful in proportioning new boilers, and comparing the efficiencies of the furnaces of different boilers, it must not supersede practical trials. When an excess of air is allowed to enter, *a* must be multiplied by the fraction—

$$\left(\frac{\text{weight of air supplied}}{\text{weight which ought to have been supplied}}\right)^2$$

The cooling effect produced in the furnace by this excess of air must be calculated (Art. 53), and proper deductions made.

* The bottoms of flues are non-effective in heating, so that the real heating-surface is about three-quarters of the total surface as measured. In our work, however, the *total exposed surface* is always to be understood as being represented by S.

Calculated Table of Efficiencies of different Furnaces belonging to the four clauses just mentioned.

$\frac{S}{F}$	Efficiencies for different classes.			
	I.	II.	III.	IV.
0·1	0·16	0·15	0·25	0·22
0·25	0·33	0·31	0·45	0·43
0·5	0·50	0·46	0·62	0·59
0·75	0·60	0·55	0·71	0·68
1·0	0·66	0·61	0·77	0·73
1·25	0·71	0·65	0·81	0·77
1·5	0·75	0·69	0·83	0.79
2·0	0·80	0·73	0·87	0·83
2·5	0·83	0·76	0·89	0·85
3·0	0·86	0·79	0·91	0·86
6·0	0·92	0·84	0·95	0·90
9·0	0·95	0·87	0·97	0·92

On examining this table, it is seen that when S is small in comparison with F, the efficiency is small, but *increases when a blast is used in the chimney*; that as S gets greater in proportion to F, the efficiency increases, its limit, of course, being unity; *that in general the forced draught is considerably more efficient than the chimney*, until when S becomes *very great* there seems to be the same efficacy from both.

Example.

Boiler of 1st class.—S = 48 sq. feet of heating surface per sq. foot of grate.

" F = 25 lb. of fuel per hour per sq. foot of grate.

Then $\frac{S}{F} = \frac{48}{25} = 1{\cdot}72$, hence from the table the efficiency is 0·78. Now, if the fuel used is coking coal, we see from Arts. 53 and 28, Book I. that the total quantity of heat from one pound of this coal is sufficient to convert $\frac{8540}{536}$ or 15·9 lb. of water at 100° into steam. In fact, the theoretic *poundage*, or *evaporative power*, is 15·9. Hence in the boiler the real poundage is 12·4. We may evidently use the efficiency just found to calculate the quantity of water converted by this furnace into steam by 1 lb. of any of the fuels in Art. 53.

Rankine, who first enunciated this law, shows by many examples that, in a striking manner, it agrees with the experiments of Clark on locomotives, and with those of the Newcastle Committee and of the Admiralty reporters, so long as F is less than 60 lb. When the combustion is so rapid that $F = 60$, it is probable that even with care it is incomplete, and the cause of loss No. III. has also to be considered.

The usual poundage of water obtained in practice from 1 lb. of coke or coal is 10·8.

In practice, the poundage is found to increase by one five-hundredth part of itself for every degree of rise in temperature of the feed-water.

(1.) The theoretical poundage is 14, $S = 54$, $F = 27$, with chimney draught, no water-heater: find the real poundage.—*Ans.* 10·22.

(2.) Find the number of units of heat usefully given out by the burning of 1 lb. of dry wood (Art. 53) in a furnace, where $S = 42$, $F = 28$, in a boiler of Class I. —*Ans.* 3000 units.

(3.) The theoretical poundage is 15·5, $S = 60$, $F = 64$. Boiler of Class II. (Rankine's Experiment).—*Ans.* 13·48.

Locomotive Boilers.

The temperature of the fire-box is as great as 1600° C. in the centre of the burning coke. The temperature of the smoke-box is usually 330° C.

The heating-surface as before = surface of fire-box above grate ÷ inside surface of tubes.

It is known that the usual tube area is equal to from 6 to 18 times the fire-box area.

The heating surface is never less than 85 square feet per square foot of grate.

The combustion is imperfect when there is more than 60 lb. of coal burnt per hour per square foot of grate, although in some engines this is as much as 112 lb.

The smallest fire-grate ever adopted is one of four square feet.

The speed of the engine seems to have no effect on the efficiency of the furnace.

In locomotives, the loss by radiation, &c., is made very small by the use of a garment of felt, pressed down with pine-battens and hoop-iron. The chimney has an outer shell, the inter-space being rendered non-conducting.

(4.) The theoretical poundage of 1 lb. of coke being 14·1, $S = 60$, $F = 56$: find the real poundage (Clark's Experiments).—*Ans.* 10·43.

(5.) Burning coke as in (4), $S = 66·4$, $F = 56·2$ (Clark). —*Ans.* 10·72.

(6.) Burning coke as in (4), $S = 60$, $F = 87$ (Clark). —*Ans.* 9·3.

CHAPTER XVI.

THE SHELL AND FLUES. BOILER EXPLOSIONS.

143. **The Capacity of Boilers Generally.**—This varies very much according to the nature of the furnace and of the engine. If the *volume of the boiler* in cubic feet be divided by the *area of the heating-surface* in square feet, we get what may be called the *mean depth* of the boiler with regard to heating-surface.

The *mean depth* in feet, as given by the practice of the best engineers, is shown in the following table :—

Egg-ended boiler, no internal flues .	3·50.
Cornish boilers	1·65 to 1·00.
Multitubular marine boilers . .	0·50.
Locomotive boilers, and others . .	0·10.

Having settled the quantity of fuel to be burnt per hour, we may use Art. 33 to determine the area of the fire-grate. From this a consideration of the table of last Art. will enable us to determine the area of heating-surface.

The *nominal horse-power* of a boiler is misleading and indeterminate, and will not here be entered into.

The area in square feet of the heating-surface, multiplied by the mean depth, as given in the above table, determines in cubic feet the volume of the boiler.

144. **The Shell.**—The shell is usually formed of plates of good wrought iron, riveted together with single or double rows of rivets. Angle-irons, or gussets, usually connect the plates at the corners, although the plates are very often rolled at the corners into flanges as a safer arrangement.

The plates are $\frac{3}{8}$ or $\frac{1}{2}$ inch thick, and are of a moderate width, breaking joint along the boiler. The flat ends are half as thick again as the cylindric parts.

No projections are allowed which might prevent free circulation of water along the plates.

Joints are either formed by riveted plates lapping in the ordinary way, by their edges being planed and butted against each other, and the seam covered by a strap or welt, or by the plates being welded together by a scarf or lap.

Welts, or straps, seem to be stronger than other riveted joints, most likely from needing fewer holes than are required in other methods of riveting. It seems likely, from experiments, that scarf-welded joints will eventually prove best.

While the tenacity of good boiler-plate is usually given as 51,000 lb. per square inch section, and that of a double-riveted joint at 51,000 lb. per square inch of section of the iron left between the rivet-holes, that of a single riveted joint is somewhat less than this, because of the unequal distribution of the stresses.

Letting the section of plates at a joint be *the whole length of the joint multiplied by the thickness of a plate*, then the tenacity per square inch at—

	lb.
A scarf-welded joint	46,000.
Double-riveted, double-welt joint, the diameter of each hole being $\frac{3}{10}$ths of the distance from centre to centre of the holes .	38,000.
Double-riveted lap-joint	35,700.
Lap-welded joint	31,400.
Double-riveted single-welt joint . . .	31,500.
Single-riveted lap-joint	31,000.
A number of single-riveted joints properly broken	34,000.

One-eighth of these tenacities is generally employed as the *working load*, and when plates are subject to galvanic action or oxidation by great heat, the working load is even made less than this.

If p is the pressure per square inch in the boiler *above that of the atmosphere*, f the *working load* per square inch as above, t the thickness of the plates in inches, and r the radius of the shell in inches, then it may be shown that—

$$p = \frac{ft}{r}$$

This equation will be found very useful in determining the working pressure in a given boiler, or the thickness of a boiler corresponding to a given working pressure.

It will be seen from this equation, that for a given working pressure the thickness of boiler-plate is greater as the radius is greater—that is, it is proportional to the radius.

In practice it is found convenient to use a thickness of $\frac{3}{8}$ths of an inch for most ordinary boilers, as this seems most favourable for good workmanship in riveting, &c. Increased strength is obtained by diminishing the radius of the shell.

145. **Strength of Flues.**—Fairbairn found that internal flues gave way by collapsing by pressures which

$$\propto \frac{t^{2.19}}{l\,r}$$

where l is the length, r the radius of the flue, and t the thickness of plates. Instead of 2·19 we generally employ 2, and when measurements are in inches, and

p is the working pressure above that of the atmosphere in pounds per square inch—

$$p = 605000 \frac{t^2}{l r}$$

for ordinary *cylindric flues* whose joints are properly broken.

Much depends on the flues being truly cylindrical, and Fairbairn recommends butt-joints and straps instead of lap-joints.

For *elliptic flues*, insead of r in the above formula, we must use—

$$\frac{a^2}{b}$$

where a is the greater, and b the smaller radius of the ellipse.

It is well, for the sake of conductivity and for other reasons, to have the thickness of the flue the same as that of the boiler-shell. If, from the above formula, the flue is found to be weak, let rings of 3-inch angle-iron be placed round the flue at equal intervals. Fairbairn found that the strength of a flue was inversely as the distance from ring to ring in such a case. One ring at least ought to be used.

It is recommended that the strengthening rings should not touch the flues, but be separated from these by ferules, placed eight inches apart. When there are two flues, the rings of different flues need not be opposite each other, as this would give corners for the gathering of incrustation.

146. As steel is becoming comparatively cheap, and its quality more uniform, steel boilers are being introduced. In general, steel boilers are as strong as wrought-iron boilers of $1\frac{3}{8}$ times the thickness. The excellent conductivity of this thin shell makes its use very desirable.

It is found that steel plates are, in general, too hard. It is recommended that more attention should be paid by manufacturers to methods for obtaining ductility and homogeneity of material. Flues of good steel plates, when carefully caulked and when annealed after being flanged, are no more apt to fail than flues of wrought-iron plates.

Joints are caulked after the riveting by means of a blunt chisel and a heavy hammer; the boiler is then proved by water-pressure, and caulked at leaky places after the water has been removed. The joints are rusted, and painted over with a thin putty, which is afterwards dried gently.

The testing-pressure of a boiler ought in no case ever to be greater than half the bursting pressure, but it ought never to be less than twice the working pressure.*

* The results of experiments conducted by the Commissioners on the Application of Iron to Railway Structures, and also those by Mr. Fairbairn, go to show that a structure may be permanently injured by a load which is much less than what is understood to be the breaking load; and the statement is often made that a *proof load* must not be applied which will give a permanent set to a structure, when this structure is not merely got up for the experiment, but is to form a part of some real engineering arrangement. Such a statement is too general, for it is evident that a riveted joint is subjected to a complexity of stresses about the rivets; and the duties of these will be very unequal, and a set in the direction of what will be the working stress may add to the strength of such a joint. Again, a bar, or plate of iron, on leaving the rolling-mill, may have such a complexity of stresses within its structure as may allow of its easily receiving a set in some direction, and thereafter refusing to set further, except for a very great additional load. Suppose a wire pulled with a force sufficient to make some part of it yield, it is evident that this part will be the weakest; and if it is additionally weakened by yielding, it will yield faster and faster till rupture ensues; but if, instead of weakening with the first slight set, the part strengthens, then the wire will yield all along its length, and every part of it will be uniformly strengthened. This latter happens to copper wire when taken in the condition in which it is usually sold, and practical bell-hangers stretch it for the purpose of strengthening it.

This subject is of importance in connection with boilers. Professor James Thomson says that the safety of a boiler consists in what we know about it, and has nothing to do with so-called *theoretical* strength.

The bricks of flues subjected to much heat are set in clay instead of in mortar. The bricks immediately supporting the boiler must be made of a peculiar shape. The mode of setting shown in our sketch of a Cornish boiler ought to be studied. Water from leakages, &c., tends to settle at points of support, and is capable of doing mischief. Joints must be kept clear of the brickwork. In lapped joints the upper plate ought to be outside. Parts of boilers otherwise exposed are covered with non-conducting felt, bound by battens, or with a layer of ashes, or with brickwork.

147. **Deposition.**—Water containing certain salts becomes less capable of holding them in solution when heated, and so deposits them; but the greater part of the deposition is due to much water going off in steam, and leaving super-saturated solutions behind. These depositions hurt the conductivity of the plates, so that much coal is wasted. Again, through the non-conductivity, the side of the plate next the furnace becomes too highly heated, is gradually corroded, and the plate is weakened.

The best remedy is that of blowing off the water at intervals as it becomes impure. Where there is little pure water to be had, surface-condensation ought to be employed in the engine, and the condensed water returned to the boiler (Art. 98).

A crust will be deposited from the smoke in the tubes and flues, and, as it is non-conducting, will seriously impair the efficiency of the furnace unless it is removed at intervals. The effect of this may be seen from the log-books of marine engineers. In one instance the coal burnt per horse-power per hour increased by

Our knowledge of a boiler's strength is obtained solely from the proof to which we have subjected it; and the ratio of the proof-pressure to the working-pressure is the only factor of safety which should be thought of.

3·6 lb. in one month, through neglect or inability to clean the tubes.

148. **Explosions.**—Explosions arise from overloading of the safety-valve, from original weakness in the boiler, from corrosion of the plates, which ought to be inspected at short intervals of time, or from the sudden generation of steam consequent on the heating of the plates, due to irregular supplies of feed-water.

This latter is, perhaps, the most common cause of explosions. We saw in Art. 32 that, when a metallic plate was at a high temperature, water lying on the metal might be separated from it by a cushion of non-conducting steam. Now, when the crown, or tubes, or top of the flues of a furnace become uncovered, they rapidly get heated; so that any water which may afterwards be thrown on them will assume the spheroidal state. Deposited scale is apt to detach itself, uncovering the heated metal with the same result. Sooner or later the metal cools below 150° C., and becomes no longer separated from the liquid; consequently there is a sudden copious production of steam sufficient to burst the boiler.

It is possible by forcing the fires to heat the metal beyond 150° without its getting uncovered with water; and after the cushion of non-conducting steam is formed the temperature may get exceedingly high. From this we see that we may reach a limit to the possible temperature of a furnace, with the present thickness of plate; so that still smaller boilers must be used either singly or combined.

It is well to guard against the fires being forced too much. When the water has by accident got too low, and the plates are uncovered, the engineer must shut off the feed-water, let the fires be raked out immediately, and let all possible valves be opened for the escape of steam and water.

CHAPTER XVII.

EFFICIENCY OF THE ENGINE, DYNAMOMETERS, AND INDICATORS.

149. CONSIDERING the boiler and cylinder and working parts as forming one engine, we may define the efficiency as the ratio of the useful work performed to the work represented by the total heat of combustion of the fuel. From Art. 53, we see that the total heat of combustion of 1 lb. of ordinary fuel varies from 8540 to 7000, and these numbers indicate 11,870,600 and 9,730,000 foot-pounds of work respectively. This is disposed of and distributed in the following manner:—

(1.) The heat which disappears uselessly in the furnace, and which varies in amount from 10 to 60 per cent. of the total heat. (2.) The heat remaining in the working-fluid, steam or water, when it is rejected. (Art 41). (3.) The heat lost by the cylinder through want of non-conductivity or other defect. (4.) The heat lost in friction, &c., by the moving parts, and (5) the useful work.

The work performed by a steam-engine is found by measuring the *resistance* and the *space* through which it is overcome. These measurements are made by dynamometers. Again, the work performed by the steam may be calculated from the curves traced out by the pencil of Watt's indicator.

150. **Dynamometers.**—Resistance may be measured in terms of the pressure of a column of water or other weight, or, in a better manner, by means of a spring. In *Morin's* instrument for obtaining *the work performed*

in traction, the load is connected with the agent by means of a powerful spring carrying a pencil, which changes its position horizontally as the resistance varies from time to time. Paper passing laterally at the speed of the load under the pencil has a curved line marked on it, whose ordinates represent resistance and space respectively. From this indication (by means of its area) the work may be calculated.

The paper cylinder derives its motion from a bearing wheel of the load, if on land; and when the resistance is that of a towed vessel, from a little vane immersed in the water indicating the speed.

Morin's instrument for measuring *the work transmitted to a revolving shaft* is similar in principle.

A pulley is fastened to a shaft by means of an elastic radial arm, firmly fixed at one end to the shaft, at the other to the rim of the pulley. This pulley transmits motion by means of a belt, the spring being more or less strained as the resistance is great or small. A pencil on the spring marks a strip of paper moving towards the centre of the pulley at a rate governed by that of the machinery, and from the resulting curve the work done may be calculated.

In *Prony's Friction Dynamometer* the resistance is measured by friction on as haft, produced by measurable pressures, the space being measured separately.

151. **The Indicator.**—The instrument which most readily enables us to find the work performed by a steam-engine is *Watt's indicator*, of which the most common modern form is that shown in Fig. 1, and whose use has already been described in part in Art. 29, in explaining the properties of steam and other fluids.

U is a cylinder bored truly, and fitted with a piston moving steam-tight, below which at any time steam, as it exists above or below the engine-piston, may be allowed to enter. The piston-rod is attached to a long spiral spring, hidden in the case S T, of which equal

changes in pressure on the piston produce equal changes in length. The pencil P in the holder O P shows by its movement up and down increase and diminution of pressure.

This is McNaught's form of the indicator; in that of Richards the spring is much stronger, and not so liable to vibrate, so that the piston has only a small amount of motion up and down, this motion being increased for the pencil by means of a lever and parallel motion.

The paper cylinder P may make about seven-eighths of a revolution when the string R is pulled, and is enabled to come back to its old position by means of a concealed spring.

R is connected with some moving part of the engine, which reciprocates with the piston; usually a radius bar of the parallel motion in beam-engines, or a temporary vibrating piece in others.

Hence, when the paper covering P is unwound, horizontal distances will represent the spaces passed over by the engine-piston at different times; and hence the volume of steam in the cylinder at different times; vertical distances representing the different pressures of the steam. When the indicator is detached, the pencil-holder is found at 14·73 on the scale M N, for the little cylinder is now connected with the atmosphere, whose ordinary pressure is 14·73 lb. per square inch. In this position, when R is pulled, the *atmospheric line* is traced on the paper. When the instrument is placed in connection with the vacuum of a condenser, the pencil-holder will be found nearly at zero of the scale. When the pencil is at *zero* and R is pulled, the *vacuum line* (seldom used in practice) is traced on the paper.

152. Let O V (Fig. 2) represent the atmospheric line when the engine is non-condensing, and let it represent the vacuum line when the engine is con-

P 2

densing. In either case the pencil never comes below this line.

Without turning the paper, let the pencil mark the vertical line O P, as the spring is moved up and down.

When B (Fig. 1) opens connection with the bottom of the engine-cylinder, and R is properly attached, the curve A B C D E (Fig. 2) will be traced on the paper as the steam does work on the piston by moving it through the forward-stroke. In the back-stroke, work is done by the piston on the steam still remaining behind it, and the curve E F A is traced out.

Distances such as K M and K N represent the pressure and volume of steam in the cylinder at a certain instant.

153. Let us examine the indication from a condens-

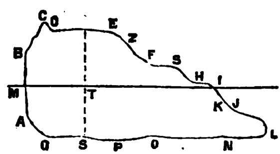


FIG. 53.

ing-engine (Fig. 53), in which M I represents the atmospheric line.

We may suppose A to indicate that the piston is at the end of the cylinder, since it is the farthest point to the left; that at present the pressure is less than that of the atmosphere, and that the steam-port is about to open, since the curve A B is nearly vertical.

It is just possible that the steam-port is not open, and that the sudden rise in pressure A B is a continuation of that shown from O to A, and further continued to C, arising, for the most part, in very great *cushioning* of the steam, or compression of the steam, behind the

piston, after communication has ceased with the condenser. An examination of the amount of *clearance* (or space left in the cylinder at the end of the stroke) will tell whether or not this is possible. In ordinary engines the configuration O A B C would mean that at O the exhaust-valve was closed, and that cushioning took place from O to M, or perhaps that the steam-port opened before A, and, as it opened gradually, the steam had not full pressure until the piston had been driven some distance. At C there was a tremor in the spring, whose indication must be disregarded. The steam begins to diminish in pressure at E, by the gradual closing of the valve giving rise to expansion through wire-drawing, or by the sudden closing of the valve giving rise to expansion of the steam already in the cylinder.

The interrupted curve from E to L may be explained by supposing vibrations in the steam-space due to rapid expansion, or to intervals of condensation due to expansion of steam at its pressure of saturation, or to vibrations of the spring. At L the piston has reached the end of the cylinder, which is now filled with steam at a very low pressure. It is evident from the appearance after this point that the exhaust-port has already been opened. L O represents the nearly uniform pressure in the space which is now virtually a part of the condenser.

The shape of a non-condensing engine diagram may be considered in the same way.

When the speed is very small, the curve is more regular, sharp turns indicating the points of admission, cut-off, and release; and when we require indications for observing the working of the tappet-motion or link-motion it is well to make the speed as small as possible.

Fig. 2 shows a locomotive diagram, taken when the engine was moving slowly. A is the point of admission, C of cut-off, D of release.

154. To find the effective work done on the piston per square inch of its area, we refer to Fig. 2. If K M is very near to L S, we see that, since K M very nearly represents the effective pressure per square inch on the piston, and M S the space through which the piston moves (representing the increase in volume of the steam), K M × M S will approximately measure, or the area K L S M will accurately measure, the work done in passing from M to S.

Drawing many lines like K M, dividing the horizontal distance from B to E into many small parts, the whole area bounded by B C E, and the line of no pressure represents the work done by the steam on the piston helping its motion, and the area contained by E F A B and the line of no pressure the work done by the steam *against* the motion of the piston. Hence the difference of these, or the area of the closed curve B C D E F A represents the effective work done by the steam on one side of the piston in one complete stroke.*

155. Now, the greatest length of the diagram measured horizontally represents the space passed through by the piston in feet, and is known since we are given the length of the crank; and as the pressure is given by the indicator-scale in pounds, the area of the figure represents the work in foot-pounds done by the steam on *one* side of the piston in one complete stroke or a revolution of the crank.

To calculate this area approximately, it is usual for engineers to measure the pressure at ten places equally distant from each other (leaving two half-spaces at the ends), finding the mean pressure from these measurements, and multiplying by the length of the stroke.

* When the engine is single-acting, one stroke is performed during every revolution of the crank; when double-acting, two strokes are performed during this time. The end of the cylinder is the limit of the forward stroke, and by the length of the stroke it is usual to understand the length of the *forward part of the complete stroke*—that is, the length of the cylinder, or, rather, twice the length of the crank.

156. Given, then, an indication to find the H.P. of a given engine. Draw at right angles to the atmospheric line ten lines equally distant from each other, leaving two half-spaces at the very extremities of the curve. Measure these chords of the curve, and divide the sum of the ten numbers obtained by ten, for the mean effective pressure of the steam on the piston. Use this pressure P in the formula $\frac{PLAN}{33000}$, where L is twice the length of the crank, A the area of the piston in square inches, and N the number of revolutions of the crank in single-acting engines, or where N is twice the number of revolutions in double-acting engines.

Strictly speaking, we ought to have a diagram from both sides of the piston at the same time, as the valves may not distribute the steam equally, treating the engine like two single-acting engines.

It must be remembered, in considering indicator-diagrams that, in consequence of *clearance*, we get the changes in volume and pressure of more steam than might appear from the position of the point of cut-off in the indication.

157. Given an indication to find the work done by *a cubic foot* of steam at the initial pressure, or pressure at cut-off. When the clearance and valves are so regulated that the cushion of steam at the end of the stroke would attain the initial pressure without admission of new steam from the boiler (Art. 177), find by the shape of the diagram the volume of steam in the cylinder, minus the volume of the clearance, at the time of cut-off; in fact, find the length of the stroke passed over at cut-off in feet, and multiply by the area of the piston in feet; divide the work done in one stroke by this product for the work done by a cubic foot of steam at the initial pressure.

To calculate the work done by *one pound of steam:* multiply the work done by 1 cubic foot by the number corre-

sponding to the initial pressure in the third column of the table of Art. 32.

When the steam is wire-drawn, the volume at the initial pressure of the quantity which has entered ought to be measured by considering the volume and pressure some time before exhaust, as the indication will there be greatest. In all cases we find the volume at the initial pressure which has entered, so that allowance must be made for condensation.

158. In determining the work done by a cubic foot of steam, we divided the work done in one stroke by the volume at the initial pressure of the steam lost during the stroke. If we had divided the work done in one stroke by the work done before expansion begins—that is, if we had divided the area of the indicator-diagram A G J C D (Fig. 9) by the area A G G′ D *—we should have determined what has lately been called the "Indicator Co-efficient" of the engine.

Evidently, the "Indicator Co-efficient" is the work done by a cubic foot of initial pressure steam just determined, divided by the initial pressure per square foot.

The "Indicator Co-efficient" will in succeeding pages be recognized under the forms—

$$r\frac{(p_{1M} - p_3)}{p_1} \text{ or } \frac{rp_e}{p_1}$$

159. In *double-cylinder* engines high-pressure steam expands above a piston in a small cylinder, and at a certain instant, instead of communication with the atmosphere being made, the steam is allowed to enter the upper part of a large condensing cylinder, where it does still more work; so that above both pistons it is greatly reduced in pressure, and is doing positive work in the large cylinder, and negative work (or it is retarding the motion of the piston) in the smaller.

It is usual to consider the work done by each engine separately, by taking an independent indication from each cylinder.

For theoretic purposes it is sometimes useful to have a dia-

* In the diagram in question there is no back-pressure shown. But it is understood that the area A G G′ D is independent of the back-pressure, in fact, that the area A G G′ D is equal to the length of the stroke before cut off, multiplied by the usual pressure.

gram representing the variations in volume and pressure of the steam, as if it expanded in one cylinder only.

It will be necessary to reduce both indications to the same scale—that is, on a certain scale the length of each indication must represent the *whole volume* of the corresponding cylinder. Both indications are placed on the same zero line.

Fig. 54 shows the indications A B K K′ K″ D F′ and F G G′ G″ D F″ placed on the same zero line O V, the greatest length measured horizontally of each indication, representing the area of the corresponding piston in square feet multiplied by the length of its stroke. A sufficient number of horizontal lines F L, F′ L′, &c., are drawn, *and the distance* K L *in every case is made equal to* F G. The points K, L, L′, L″, &c., deter-

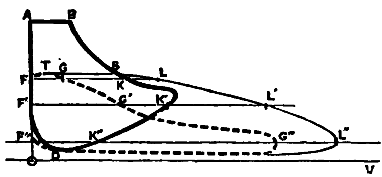


FIG. 54.

mine a new diagram A B S L L′ L″ D F′ which represents the variation in volume and pressure of the steam as if it expanded in one cylinder only.

160. Gooch's Indicator (Fig. 55), which is less liable to oscillations of the spring at high speeds, has steam admitted from *b* to one side of a small piston *a* by means of a slide-valve. The piston works horizontally, its rod compressing an elliptical spring, and also working the small arm of a lever carrying a pencil at its extremity. This pencil moves in arcs of circles. A long strip of paper unwinds beneath the pencil at a rate proportional to the speed of the engine; so that the diagram never forms a closed curve, and indications are given of a great many strokes in one long line. A second pencil fixed to the frame of the indicator continually traces out an atmospheric line. These diagrams may easily be reduced to the usual form by the help of a

scale curved to correspond with the arc of the circle in which the pencil moves.

161. M. Deprez, employs below the ordinary indicator spring, a screw, by means of which the steam is prevented from lifting the piston until the pressure exceeds a certain amount. Until this instant the indicator-diagram is a horizontal line; it now becomes curved. When the screw is turned the piston is again prevented from moving until the pressure exceeds a certain other limit, so that a

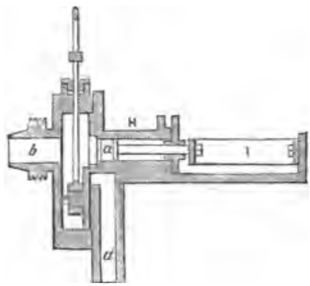


FIG. 55.

series of corners is obtained, which are points on the real indicator diagram, and may be joined by hand.

When vibrations of the spring are in this way destroyed, exactly the same indication may often be obtained during four or five successive strokes.

162. The indicator enables us to discover faults in the valve-motion and in the size of steam-passages. By it, in factory-engines, we may calculate the power required for different machines under different conditions of load, speed, lubrication, &c. In determining the power required for a certain process, it is usual to take diagrams (1) without and (2) with the particular machinery in motion. Many indications ought to be taken from both ends of the cylinder during a time of observation. There is a great necessity for a delicate indicator, giving continuous diagrams.

Indicators may with advantage be applied to the condenser, air-pump, and hot-well, to determine the pressures at different parts of the stroke. In making experiments on efficiency, care must be taken to distinguish between the performances of the boiler and the engine. At the beginning and end of a series of observations we ought to have steam up, and a perfectly clear grate.

CHAPTER XVIII.

EFFICIENCY OF THE ENGINE (*continued*).

163. IT is easy to determine by the integral calculus the form of the expression for the area of the indicator diagram or the effective work done by the steam on the piston in one stroke.

If v_1 is the volume in cubic feet occupied by the steam at the beginning, and v_2 the total volume swept through by the piston during the stroke, and if p represents the difference in the pressures in pounds per square foot at similar instants in the fore and back parts of the stroke, then $\int_{v_1}^{v_2} p\, dv$ is the expression for the foot-pounds of work done by the steam on the piston in one stroke.

164. The volume v is often represented by an equivalent length in feet of the cylinder, and p is generally given in pounds per square inch, and in this case the integral represents the foot-pounds of work done during the stroke per square inch of the area of the piston; so that if a is the area of the piston, $a\int_{l_1}^{l_2} p\, dl$ is the work done in one stroke.

164*a*. From the last article it will be seen that ordinary indicator-diagrams are not formed of mathematical curves, their peculiarities arising (1) from the irregularity in conduction of the cylinder sides, (2) from tremors in the piston and connections, (3) from tremors, friction, and sluggishness of the spring in the indicator.

Hence, in calculations regarding the work done by steam on an imaginary piston, we first suppose that there is no clearance, no change of pressure during that part of the forward-stroke occupied by admission, nor in the return-stroke, and that at the beginning and end of the stroke we have vertical lines representing sudden changes of pressure; such indications may be examined in Fig. 9.

Theoretic indications of this kind are found to differ very little in area from the indications obtained from real engines.

165. One of the following five suppositions * must also be adopted :—

I. That the steam expands like an ordinary gas, the curve being a rectangular hyperbola, whose equation is pv = a constant (*see* Art. 33) or—

$$p \propto v^{-1}$$

This was one of De Pambour's suppositions.

In ordinary engines, when we suppose the steam to contain some liquid matter on its entrance, and that this is evaporated later on in the stroke by heat obtained from a steam-jacket, the supposition that during the expansion p varies as v^{-1} is approximately correct. It has been found by experiment that with

* This method of treatment is due to Rankine and Clausius, who obtained similar expressions independently, for the work done in a cylinder and the amount of steam condensed. Rankine discovered, by means of actual drawings of the curves, the expressions of the form $p \propto v^{-\kappa}$ for II., III., and V.

this ***nearly dry steam*** the work wasted in allowing the steam to escape at the pressure of release in condensing-engines, is—

$$15\, p_2\, v$$

foot-pounds, where v is the volume of steam released and p_2 is the absolute pressure (a vacuum is supposed to be represented by 0) of release. In non-condensing engines $14\, p_2\, v$ foot-pounds are lost.

II. That the curve of expansion is an ***adiabatic*** curve (*see* Arts. 34 and 61). This is really the case when the cylinder is non-conducting, or is well covered with non-conducting material, so that the steam never receives nor gives out heat. As was stated in Art. 61, there is condensation in this cylinder.

The approximate equation to the curve of expansion is given by—

$$p \propto v^{-\frac{10}{9}}$$

III. That the cylinder has a steam-jacket, which gives out just so much heat as prevents condensation; hence the equation to the curve connects the pressure of saturation at different temperatures with the volume in such circumstances of a given weight of steam.

The approximate equation to the curve of expansion is given by—

$$p \propto v^{-\frac{17}{16}}$$

This is often called *dry saturated steam.*

IV. That the steam is *superheated, a constant temperature* being maintained in the cylinder by means of a steam-jacket. Here the curve of expansion will be that of an equilateral hyperbola, as in I.

V. That the steam is superheated, the cylinder being well covered and non-conducting, so

that the expansion curve is adiabatic. The appropriate equation to the curve of expansion is given by

$$p \propto v^{-\frac{13}{10}}$$

166. In all the following calculations let us suppose the steam to be cut off at $\frac{1}{r}$ of the stroke :—

p_1 and p_3 the initial pressure and back-pressure respectively in the cylinder. These pressures are absolute—that is, a vacuum is denoted by $p=o$.

p_e the mean effective pressure in the cylinder.

p_m, the mean pressure during the forward-stroke; hence $p_m = p_e + p_3$.

v_1 and v_2, the volumes of steam in the cylinder at the beginning and end of the stroke.

M and H, expressions whose values depend on the nature of the engine, and which will be explained presently.

D, the weight in pounds of a cubic foot of steam at a certain pressure. This may be found for saturated steam from the table of Art. 32. Instead of referring to the table, we may employ the empirical formula—

$$D = \frac{p^{\frac{16}{17}}}{26 \cdot 36}.$$

For superheated steam $D = \frac{\cdot 936\, p}{273 + t}$ where p and t are the pressure and temperature respectively.

U, the work done on the piston by every pound of steam admitted.

a, the area of the piston in square inches.

l, the length of the forward-stroke in feet is usually called the length of the stroke, and is twice the length of the crank.

V, the mean velocity of the piston in feet per minute.
H.P., the indicated H.P. of the engine.
W, the weight in pounds of coals used per minute.
h, the useful heat in foot-pounds given out by 1 lb. of coal. In a furnace, whose poundage (Art. 142) or evaporative power is 7·24, h = 5,400,000 foot-pounds. By evaporative power, we mean the number of pounds of water converted into steam by 1 lb. of coal.
E, the evaporation in cubic feet at the initial pressure p_1 of steam *per minute.*
E_1 the evaporation in pounds per minute.

In ordinary compound engines r becomes the ratio of the volume of the larger cylinder to the volume of steam admitted at every stroke. The length of the stroke will seldom be spoken about, but it may be taken as the length of stroke of the larger piston; a the area of larger piston; V the mean velocity of the larger piston. When we come to speak of clearance we shall find c in compound engines to be the ratio of the volume of the steam-passages between the cylinders and clearance in both, to the whole volume swept through by the larger piston every stroke $l\,a$.

167. The equation connecting pressure and volume in the expansion curves of I. and IV. is given by—

$$p \propto v^{-1}.$$

The integral of Art. 165 gives the foot-pounds of work done in one stroke to be—

$$a\,l\,(p\,\mathrm{M} - p_3) \text{ where } \mathrm{M} = \frac{1 + \log. r}{r}*$$

168. Comparing the equations to the expansion-curves for II., III., and V., we see that they are generally of the shape—

$$p \propto v^{-\kappa}.$$

Where κ has different values in the five cases. The

* The logarithms here used are *Napierian*, or *Hyperbolic*.

integral of Art. 165 gives the foot-pounds of work done in one stroke in these three cases to be—

$$a\,l\,(p_1\,\mathrm{M} - p_3) \text{ where } \mathrm{M} = \frac{\kappa r^{-1} - r^{-\kappa}}{\kappa - 1}$$

169. Thus we find that in all five cases the work done in one stroke is—

$$a\,l\,(p_1\,\mathrm{M} - p_3) \quad . \; . \; . \quad (1)$$

Where in Case I. . . . . $\mathrm{M} = \dfrac{1 + \log. r}{r}$

Case II. . . . $\mathrm{M} = 10 r^{-1} - 9 r^{-\frac{10}{9}}$

Case III. . . . $\mathrm{M} = 17 r^{-1} - 16 r^{-\frac{17}{16}}$

Case IV. . . . $\mathrm{M} = \dfrac{1 + \log r}{r}$.

Case V. . . . $\mathrm{M} = \dfrac{13 r^{-1} - 10 r^{-\frac{13}{10}}}{3}$

170. Now the work of the stroke is given out by $\dfrac{a\,l}{144\,r}$ cubic feet of steam; hence, the work done by *one cubic foot* of steam at the initial pressure p_1 is—

$$144\,r\,(p_1\,\mathrm{M} - p_3) \quad . \; . \; . \quad (2)$$

The work done by *one pound* of steam at the pressure p_1 is—

$$\frac{144\,r}{\mathrm{D}}\,(p_1\,\mathrm{M} - p_3) \quad . \; . \; . \quad (3)$$

p_m or the mean forward pressure is $p_1\,\mathrm{M}$ . . . (4)
p_e or the mean effective pressure is $p_1\,\mathrm{M} - p_3$ . . . (5)

171. Again, knowing the circumstances of the different cases, it is easy to calculate from the principles

of Book I. the heat expended per cubic foot of steam admitted. Let this be represented by H in *foot-pounds* approximately, then in the different cases—

I.—$H = 144\,p_1\,(Mr + 15)$ in condensing engines.

$H = 144\,p_1\,(Mr + 14)$ in non-condensing engines.

II.—$H = 1920\,p_1 + 4000.$

III.—$H = 2230\,p_1$

IV.—$H = 590395\,D + p_1\,(3\tfrac{1}{8} + rM)$

When D is supposed to be equal to $\dfrac{\cdot 936\,p}{273 + t}$ (Art. 116) this expression becomes

$$= p_1\left(\frac{552480}{273 + t} + rM + 3\tfrac{1}{8}\right)$$

V.—$H = 773372\,D + 668\,t$

$$= \frac{723875\,p_1}{273 + t} + 668\,t \text{ or by Art. 166.}$$

In engines using superheated steam, the heat given to the steam after it leaves the boiler would otherwise most probably have been wasted, so that the increased efficiency gained by superheating is often much greater than might appear from these formulæ.

172. Now, in every case the *efficiency* of the engine will be represented by the expression—

$$\frac{\text{work done by a cubic foot of steam}}{H}$$

Hence

$$\text{Efficiency} = \frac{144\,r\,p_e}{H} \text{ or } \frac{144\,r\,(p_1 M - p_3)}{H} \quad \ldots (6)$$

The *weight of feed-water* required per cubic foot of steam expended is D.

The *weight of injection-water* required per cubic foot of steam admitted is—

$$\frac{H - 144\,r\,(p_1 M - p_3)}{1390\,r\,(T' - T'')} \quad \ldots (7)$$

where T′ is the temperature of the water in the condenser, and T″ that of the injection water.

$$\text{Evidently } V = \frac{33000 \times \text{H.P.}}{a\,p_e} \quad \ldots \; (8)$$

$$\text{H.P.} = \frac{a\,p_e\,V}{33000} \quad \ldots \; (9)$$

As there are $\frac{l\,a}{144\,r}$ cubic feet of steam used every stroke, E the evaporation in cubic feet of steam per minute at the initial pressure $= \frac{a\,V}{144\,r}$ and E_1 or the evaporation in pounds per minute

$$= \frac{D\,a\,V}{144\,r} \quad \ldots \; (10)$$

It must be remembered that p_1 must be known before D can be found.

$$\text{Evidently H.P.} = \frac{144\,r\,(p_1\,M - p_3)}{33000} \times E$$

Or

$$= \frac{144\,r\,p_1\,M - p_3}{33000} \times \frac{E^1}{D} \quad \ldots \; (11)$$

Again

$$\text{H.P.} = \frac{W \times \text{efficiency}}{\cdot 00611} \quad \ldots \; (12)$$

When 7·24 is used as the evaporative power of the boiler. For other evaporative powers of the boiler, instead of ·00611, we must use $\frac{\cdot 04424}{\text{new evaporative power}}$.

173. To assist in the use of the above formulæ, the following table has been calculated.* The value of M is given for all sorts of engines, as the period of cut-off varies from $\frac{1}{20}$th to $\frac{9}{10}$ths of the stroke. Columns I. IV. and are identical :—

* Rankine, pp. 442—3, 568.

Giving M, *or* $\frac{p_m}{p_1}$, *according to the five conditions for different values of r.*

	I.	II.	III.	IV.	V.
r	$M=\frac{1+\log r}{r}$	$M=10r^{-1}-9r^{-\frac{10}{9}}$	$M=17r^{-1}-16r^{-\frac{17}{16}}$	$M=\frac{1+\log r}{r}$	$M=\frac{13r^{-1}-10r^{-\frac{13}{10}}}{3}$
20	·200	·177	·186	·200	·149
$13\frac{1}{3}$	·269	·244	·254	·269	·210
10	·330	·303	·314	·330	·266
8	·385	·356	·370	·385	·318
$6\frac{2}{3}$	·435	·407	·417	·435	·367
5	·522	·496	·505	·522	·456
4	·596	·572	·582	·596	·534
$3\frac{1}{3}$	·661	·639	·648	·661	·603
$2\frac{6}{7}$	·717	·697	·707	·717	·665
$2\frac{1}{2}$	·765	·748	·756	·765	·720
$2\frac{2}{9}$	·809	·797	·800	·809	·770
2	·846	·833	·840	·846	·813
$1\frac{9}{11}$	·878	·869	·874	·878	·851
$1\frac{2}{3}$	·906	·894	·900	·906	·884
$1\frac{7}{13}$	·929	·923	·929	·929	·913
$1\frac{3}{7}$	·950	·945	·945	·950	·937
$1\frac{1}{3}$	·965	·960	·960	·965	·957
$1\frac{1}{4}$	·978	·976	·976	·978	·973
$1\frac{3}{17}$	·989	·986	·986	·989	·985
$1\frac{1}{9}$	·995	·997	·997	·995	·993

Examples.

(1.) In an ordinary condensing-engine, where no provisions are made for keeping the steam dry during expansion, let the initial pressure, or $p_1 = 34$ lb. per square inch; mean back-pressure $= 4$ lb. per square inch; and let the cut-off be at one-fifth of the stroke, that is, let $r = 5$. Suppose the engine to belong to Class I.

From the table, M is found to be ·522. Mean effective pressure $= p_1\,\text{M} - p_3 = 13{\cdot}75$ lb. per square inch. Work done by 1 cubic foot of steam $= 144 \times 5 \times (p_1\,\text{M} - p_3) = 9890$ foot-pounds.

$$\text{Efficiency} = \frac{9890}{144 \times 5\,(17{\cdot}75 + 15 \times 6{\cdot}8)} = {\cdot}115$$

(2.) To the data of Example (1), let us add that the evaporation is 800 feet of steam per minute. From (12) the indicated H.P. $= \dfrac{9890 \times 800}{33000} = 240.$

(3) In a condensing-engine, belonging to Class II. and, therefore, provided with a non-conducting cylinder, let there be 1880 cubic feet of steam used in one minute. Let the initial pressure, or $p_1 = 30$ lb. per square inch; mean back-pressure $p_3 = 2{\cdot}3$ lb. per square inch; and let the cut-off be at one-fifth of the stroke $\therefore r = 5$.

From the table, M is found to be ·496 : hence, from Formula (2.) the work done by one cubic foot of steam is—

$$144 \times 5\,(30 \times {\cdot}496 - 2{\cdot}3) = 9057{\cdot}6 \text{ foot-pounds.}$$

$$\text{The efficiency} = \frac{9057{\cdot}6}{1920 \times 30 + 4000} = {\cdot}147.$$

From (12) the indicated H.P. $= \dfrac{9057{\cdot}6 \times 1080}{33000} = 297.$

(4.) Data as in Examples (1) and (2), but let the cylinder have a steam-jacket, and belong to Class III.

From the table, M is found to be ·505; hence, p_e, or $p_1 \text{M} - p_3 = 13{\cdot}17$ lb. per square inch.

$$\text{Work done by 1 cubic foot of steam} = 144 \times 5 \times 13{\cdot}17 = 9482{\cdot}4 \text{ foot-pounds.}$$

$$\text{Efficiency} = \frac{9482{\cdot}4}{2230 \times 34} = {\cdot}125$$

$$\text{The indicated H.P.} = \frac{9482{\cdot}4 \times 800}{33000} = 229{\cdot}8$$

It is to be understood that clearance has not yet been considered, and that the following rules are only approximately correct.

(5.) To find the mean velocity of the piston, when the consumption of steam is just equal to its production, that is, when p_1 is the pressure of steam in the boiler. From (10) we have $\text{E}^1 = \dfrac{\text{D}\,a\,\text{V}}{144\,r}$ from which

$$\text{V} = \frac{144\,r\,\text{E}^1}{\text{D}\,a}$$

By means of this formula, given the evaporation in pounds per minute, the area of the piston, the cut-off, and given the pressure in the boiler so that D may be found, we readily calculate V, called the velocity of maximum effect.

(6.) In the last Example, when given the coal burnt per minute instead of the evaporation, from (12), after simplification, we get—

$$\text{V} = \frac{4752000\,r\,\text{W}}{a\,\text{H} \times {\cdot}00611}$$

when the evaporative power is 7·24 lb. of water per 1 lb. of coal. Simplifying, this becomes—

$$\text{V} = \frac{777{,}740{,}000\, r\, \text{W}}{a\, \text{H}}$$

H may be found from Art. 171, when the boiler pressure is known.

For other evaporative powers we use—

$$\text{V} = \frac{777{,}740{,}000\, r\, \text{W}}{a\, \text{H}} \times \frac{\text{new evaporative power}}{7{\cdot}24}$$

(7.) To find the H.P., given the evaporation in pounds per minute, the velocity in feet per second of the piston, the area of the piston, and the cut-off. Using (10) and (9), find D from the formula—

$$\text{D} = \frac{144\, r\, \text{E}^1}{a\, \text{V}}$$

By means of Art. 166 given D find the corresponding pressure p_1, and the indicated horses' power—

$$\text{H.P.} = \frac{a\, \text{V}\, (p_1\, \text{M} - p_3)}{33000}$$

(8.) To find the H.P., as in last example, given the weight of coals burnt per minute, and that 1 lb. of coal gives out h = 5400000 foot-pounds of energy usefully in the furnace. Here W h × efficiency = V p_e a, or—

$$\text{W}\, h\, \frac{144\, r\, p_e}{\text{H}} = \text{V}\, p_e\, a$$

Or

$$\frac{144\, r\, h\, \text{W}}{\text{H}} = \text{V}\, a$$

Again, from (12)

$$\text{H.P.} = \frac{144\, r\, \text{W}\, (p_1\, \text{M} - p_3)}{\cdot 00611\, \text{H}}$$

The student will find from Art. 171, the expression of H in terms of p_1, and will then eliminate p_1 from the above equations.

(9.) To find the evaporation in pounds per minute, given the indicated H.P., V, a, and r. Transforming the equations of Example 7, we get the rule: find p_1 from the following—

$$p_1 = \frac{33000 \times \text{H.P.} + a\, \text{V}\, p_3}{a\, \text{V}\, \text{M}}$$

Find D (Art. 166) by means of p_1, and the evaporation in pounds per minute—

$$\text{E}^1 = \frac{a\, \text{V}\, \text{D}}{144\, r}$$

(10.) To find the weight of coals burnt per minute.

The equation found in Example (8) is easily transformed.

(11.) To find the initial pressure in the cylinder, knowing the evaporation E^1 in pounds per minute. From the second equation of Example (9), find D, and thence find p_1.

(12.) To find the initial pressure, knowing the weight of coals W burnt per minute, V, a, and r. By means of the second equation of Example (8), and $\text{V}\, a\, p_e = 33000 \times \text{H.P.}$, we find—

$$\text{H} = \frac{4752000\, \text{W}\, r}{\cdot 00611\, \text{V}\, a}$$

H is expressed in terms of p_1 in Art. 171, so that p_1 may be calculated.

Evidently, for other evaporative powers of the boiler than 7·24, instead of ·00611 we may use—

$$\frac{\cdot 04424}{\text{new evaporative power}}.$$

(13.) To find a, the area of the cylinder in inches, given E^1, V, H.P. and r.

Using the equations of Example (7), express D in terms of p_1, and from the equation $D = \frac{144\, r\, E^1}{a\, V}$ and $H.P. = \frac{a\, V\, (p_1\, M - p_3)}{33000}$ eliminate p_1.

By means of the resulting equation we can readily find p_1.

(14.) To find a, the area of the cylinder in inches given W, V, H.P., and r.

Using the equations of Example (8), express H in terms of p_1 by means of Art. 171, and eliminate p_1 between—

$$\frac{144\, r\, h}{H} = V\, a$$

and

$$H.P. = \frac{144\, r\, W\, (p_1\, M - p_3)}{\cdot 00611\, H}$$

(15.) To find E^1, given V, a, and p_1, use Formula (10), Art. 172.

(16.) To find W, given V, a, and p_1. This is given by transforming the equations in Example (6).

$$W = \frac{a\, H\, V}{777{,}740{,}000\, r} \text{ when H is found from } p_1.$$

When the evaporative power is other than 7·24, this equation becomes—

$$W = \frac{a\,H\,V}{777{,}740{,}000\,r} \times \frac{7{\cdot}24}{\text{new evaporative power}}$$

174. By means of the formulæ given above the following table has been calculated. Students are advised to arrange similar tables for themselves, to suit the different classes of engines. These tables prove to be very useful in practice. With a set of such tables for all classes of engines, the weight of coals burnt per hour for each H.P. may at once be approximately determined in the manner shown below :—

Efficiencies of Condensing and Non-condensing Engines provided with steam-jackets, according to **Method III.** *of the preceding pages. Hence the steam may be called dry saturated steam. r and the entering pressure p_1 are supposed to be known, and the back-pressure is assigned to be 4 lb. in all cases.*

CONDENSING.

p_1	Values of r							
	10	5	3·33	2·5	2	1·66	1·25	1
20			·095	·090	·083	·075	·0625	·052
40		·131	·118	·106	·095	·086	·071	·058
60	·159	·140	·125	·111	·100	·090	·073	·060
80	·170	·147	·128	·114	·102	·091	·074	·061

NON-CONDENSING.

p_1	Values of r							
	10	5	3·33	2·5	2	1·66	1·25	1
60				·074	·070	·064	·055	·045
80			·091	·086	·080	·073	·061	·050
100		·105	·100	·093	·085	·077	·064	·053
120		·115	·107	·098	·089	·081	·067	·055
160		·127	·115	·104	·094	·085	·070	·057

175. The weight in pounds of coal burnt *per minute* for every horse's power is found by dividing ·00611 by the efficiency, if the evaporative power of the boiler is 7·24 lb. of water per lb. of coal. Instead of ·00611, we must use $\frac{\cdot 04424}{\text{new evaporative power}}$ for other evaporative powers: the efficiency is multiplied by

$$\frac{11}{30} \times \frac{7\cdot 24}{\text{new evaporative power}}.$$

176. In the following pages:—

c = clearance as a fraction of the stroke (Art. 177).

$\frac{1}{r^1}$ is the fraction of the stroke performed when steam is cut off.

r is calculated, being equal to $\frac{r^1 + c\,r^1}{1 + c\,r^1}$.

H.P. is the *useful* horse-power performed by the engine.

$$\left.\begin{array}{ll} R_1 = & \text{useful load on the engine} \\ R_0 = & \text{resistance of the unloaded engine} \\ R\ = & \text{total resistance of loaded engine} \end{array}\right\} \text{These are supposed to act at the end of the piston-rod.}$$

177. Clearance is the whole space between the piston at the end of the stroke and the valves.

When the **clearance** *and valves are so regulated that the cushion of steam at the end of the stroke would attain the initial pressure without the admission of new steam from the boiler, we may entirely dispense with the consideration of clearance in calculating the* **efficiency** *of an engine*, or in calculating the work done by a certain quantity of steam, although it affects the mean pressure during the stroke and the velocity of the piston.

When the exhaust-port is closed before the end of the stroke, by the amount $c\, r^1 \left(\frac{p_3}{p_1}\right)^{\frac{9}{10}}$. the steam-cushion will usually have the initial pressure p_1 at the end of the stroke.*

Clearance is usually expressed in a fraction as its relation to the space passed through by the piston in one stroke. This fraction we have called c, then the length of the cylinder equivalent in volume to the clearance $= c\,l$. In practice, c varies in value from $\frac{1}{8}$th to $\frac{1}{40}$th and $c\,l$ is from 2 to 3 inches in all cylinders.

As usually given, steam is said to be cut off at $\frac{1}{r^1}$th of the stroke, so that in calculating p_e, p_m, &c. (Art. 172), we must now consider the steam to be cut off when $\frac{1}{r}$th of the cylinder is filled with steam at the given initial pressure, where $r = \frac{r^1 + c\,r^1}{1 + c\,r^1}$.

* This may be worked by the student as an exercise on Book I.

Using r, we should find that the calculated mean-pressures p_e and p_m, would be too great, since we have supposed the pressure p_1 to act effectively on the piston through the space $c\,l$. In fact, if p_e is the hypothetical mean effective pressure, calculated by means of r, the *real mean effective pressure* will be $\frac{r p_e}{r^1}$

The work performed by one cubic foot of steam is $144\, r^1 \frac{r p_e}{r^1}$ or $144\, r\, p_e$ the same as if there were no clearance, the cut off being at $\frac{1}{r}$ of the stroke

177(a). Hence, when the closing of the exhaust-port is adjusted as shown above, and given that steam is cut off at $\frac{1}{r^1}$ of the stroke, let $r = \frac{r^1 + c r^1}{1 + c r}$ and from this calculate the work done by one cubic foot, the efficiency the horse-power corresponding to a certain evaporation, &c., from the formulæ of Arts. 170—3, with the help of the examples just given.

178. **Useful Work.**—Without including the back-pressure of the steam, the resistance of the engine is due to friction in the mechanism and to power needed in working the pumps. Let all resistances be supposed to act at the end of the piston-rod,* and let R_1 be the *useful* load on the engine. Then it has been found from experiment that the resistance due to friction in the engine itself is equal to the resistance R_0 in the unloaded engine, together with a constant fraction f of the useful load. Hence the total load—

$$R = R_1\ (1 + f) + R_0 \ .\ .\ .\ (1)$$

* That is, if the real resistance is k, and the velocity with which it is overcome is S; and if V is the velocity of the piston-rod, the resistance acting at the piston-rod equivalent to k will be $\frac{kS}{V}$.

Now $\frac{R}{a}$ = *mean effective pressure of steam on the piston, and may be calculated* when R_1, f and R_0 are known. Experiment shows that the unloaded resistance is equal to about 1 lb. per square inch, or $\frac{R_0}{a}$ = 1 lb., and f is usually $\frac{1}{7}$. So that the total resistance, or R in pounds $= 1\frac{1}{7} R_1 + a$.

The efficiency of the mechanism is of course

$$\frac{\text{useful load}}{\text{total load}} = \frac{R_1}{R_1 (1 + f) + R_0}$$

$$= \frac{R_1}{a \times \text{effective pressure of steam } p_e} \quad . \ . \ . \ (2)$$

In many cases in practice the total resistance may approximately be taken as $R_1 \times 1{\cdot}2$, or $R_1 \times 1{\cdot}25$.

In marine steam-engines the total resistance $= R_1 \times 1{\cdot}6$, or $= R_1 \times 1{\cdot}67$, in consequence of the motion given to the water.

Sometimes R_1 is given in *H.P.* usefully performed by the engine with a given velocity V of piston. Here $R_1 = \frac{\text{useful } H.P.}{V}$ and the indicated H.P., or V R = useful *H.P.* $(1 + f) + \frac{V R_0}{33000}$. This usually becomes

$$\text{indicated H.P.} = \text{useful } H.P. \times 1\tfrac{1}{7} + \frac{V a}{33000} \quad . \ . \ . \ (3)$$

By means of the formulæ in the last article, when given the useful load on the engine R_1, we can find the total load R, and hence, the required mean effective pressure of the steam, or $\frac{R}{a}$.

Taking clearance into account, the r^1 given for the cut off must be used to calculate r, which is equal to $\frac{r^1 + C r^1}{1 + C r^1}$. Now, the real mean effective pressure is $\frac{R}{a}$

hence $p_e = \frac{r^1 R}{r a}$ where p_e is calculated from r. To find p_1 we have $p_1 = \frac{p_e + p_3}{M}$ or

$$p_1 \times \frac{1}{M}\left(\frac{r^1 R}{r a} + p_3\right) \quad . \; . \; . \; (4)$$

M may be found from the table, Art. 173, by means of r, so that p_1 is known; hence, also the efficiency of the engine is known. Again, the work done by every *cubic foot* of steam is known, and also the work done by every *pound* of steam.

179. The Examples (5 − 16) given above now take the following shapes :—

Use in the formulæ of (5), (6), (12), and (16), instead of r the expression $\frac{r^1 + c r^1}{1 + c r^1}$.

(7.) The two equations given now become

$$D = \frac{144 E^1}{a V} \quad \frac{r^1 + c r^1}{1 + c r^1}$$

and

$$H.P = \frac{a V (p_1 M - p_3)}{33000} \times \frac{1 + c}{1 + c r^1}.$$

(8.) The two equations given, now become

$$\frac{144 h}{H} \cdot \frac{r^1 + c r^1}{1 + c r^1} = V a$$

and

$$H.P. = \frac{144 W (p_1 M - p_3)}{0{\cdot}0611 H} r^1 \left(\frac{1 + c}{1 + c r^1}\right)^2$$

(9.) In the last equation r becomes

$$\frac{r^1 + c r^1}{1 + c r}$$

(13.) The equations become

$$D = \frac{144\,E^1}{a\,V} \cdot \frac{r^1 + c\,r^1}{1 + c\,r^1}$$

and

$$\text{H.P.} = \frac{a\,V\,(p_1\,M - p_3)}{33000} \times \frac{r^1 + c\,r^1}{1 + c\,r^1}$$

(14.) The equations become

$$\frac{144\,h\,W}{H} \cdot \frac{r^1 + c\,r^1}{1 + c\,r} = V\,a$$

and

$$\text{H.P.} = \frac{144\,VV(p_1\,M - p_3)}{\cdot 00611\,H} \cdot r^1 \left(\frac{r^1 + c\,r^1}{1 + c\,r^1}\right)^2$$

In these examples, indicated H.P. alone is considered. The student will remember the formula (3) of Art. 178.

$$\text{H.P.} = \tfrac{8}{7}\,\text{useful H.P.} + \frac{V\,a}{33000}$$

180. For students who wish for approximate ready methods of working Examples 5—16, the following five very common rules are given :—

I. *To find the velocity corresponding to the maximum useful effect.*

To the logarithm of the pressure per square inch in the boiler add 0·569982, and find the natural number corresponding to the sum. Add 4·227 to this, find the logarithm, and add to it twice the diameter of the piston in feet.

To the logarithm of the evaporation in pounds per square inch add 5·08372 and the number in the second column corresponding to r^1 in the following table, and from the sum subtract the above result for the logarithm of the velocity required.

II. *To find the useful* H.P., *knowing the evaporation, velocity per minute, and area of the piston.*

To the logarithm of the evaporation in pounds per minute add 4·7364 and the number in the third column corresponding to r^1 in the following table: find the natural number corresponding to this sum.

To the logarithm of the velocity of the piston in feet per minute, add twice the diameter of the cylinder in feet and 2·7747750, and find the corresponding natural number. Subtract this result from the first, and divide by 33000 for the useful *H.P.*

III. *To find the evaporation in pounds per minute given the useful* H.P., *and the diameter and velocity of the piston.*

Find the number in the third column of the following table, and add to it 4·7363904.

To twice the logarithm of the diameter in feet, and the logarithm of the velocity of the piston in feet, add 2·74775. To the natural number corresponding to this sum, add the useful *H.P.* multiplied by 33000, and find the logarithm. From this subtract the last result for the logarithm of the evaporation in pounds per minute.

IV. *To find the initial pressure in the cylinder, given the evaporation in pounds per minute, and the velocity and diameter of the piston.*

To the logarithm of the evaporation add 3·309724 and the number in the second column of the following table, and subtract twice the logarithm of the diameter of the piston in feet, and the logarithm of the velocity. Find the natural number, and subtract 4·227. Find the logarithm of the difference, subtract ·569982, and the natural number corresponding will be the initial pressure required.

V. *To find the diameter of the piston, given the velocity in feet per minute and the evaporation in pounds per minute.*

To the logarithm of the evaporation add 47·3629 and the number in the third column of the following table. From the corresponding natural number subtract the useful H.P. × 33000, and find the logarithm of the difference. From this subtract 2·74775 and the logarithm of the velocity, and divide by 2 for the logarithm of the diameter of the piston in feet.

Table used in approximate determinations of the above five rules.

r_1	log. r	$\log\left\{\frac{r}{r_1} + \log\left(r + \frac{cl}{r}\right)\right\}$
10	·823930	·417139
8·3	·769525	·402433
6·6	·698970	·381656
5·0	·602060	·349277
4·0	·522835	·319106
3·3	·455910	·291147
2·5	·346744	·239550
2·0	·259594	·191730
1·5	·148911	·120903
1·3	·096910	·082785
1·25	·070776	·061452
1·1	·022428	·019947

Miscellaneous Exercises on Efficiency.

(1.) Find the "indicator co-efficient" of Art. 158 given initial pressure 25 lb., back-pressure $2\frac{1}{2}$ lb., cut-off being at one-fourth of the stroke, and, hence, $r = 4$. Engine of Class II. Referring to table in Art. 173, M = ·572, ∴ the co-efficient, or—

$$\frac{r\,(p_1\,\mathrm{M} - p_2)}{p_1} \text{ is } 1·704.\text{—}Ans.$$

R

(2.) By the help of logarithmic tables show that in engines of—

Class III., when $r = 5$, $M = \cdot505$.
Class I., when $r = 8$, $M = \cdot370$.

(3.) What is the work done by 1 lb. of steam, p_1 being 36, $r = 5$, $p_3 = 2\cdot5$, in an engine of Class II.? If this steam is superheated, and at 310° C., in an engine of Class V., what is the work done by 1 lb. of steam? Find the differences between your answers, and compare this work in heat-units with the heat required for superheating.

(4.) Find the heat (in foot-pounds) expended on 20 cubic feet of steam in a condensing-engine of Class I., $p_1 = 30$ lb., $r = 5$.

Find the work performed by this steam.

(5.) If the steam of Example 4 had been superheated to 300° C., and employed in engines IV. and V., what would have been the two results of answers?

(6.) How much feed-water is required to supply the place of 30 cubic feet of superheated steam at 310° C., and 36 lb. pressure?

(7.) Find the velocity of maximum useful effect when the area of the piston is 1250 square inches, pressure in boiler 44 lb., when steam is cut off at $\frac{1}{5}$th of the stroke, and when 60 lb. of steam are evaporated per minute.

$$V = \frac{144 \times 5 \times 60}{1250 \times \frac{\cdot264}{608}} = 337\cdot5 \text{ feet per minute.}$$

(8.) If, with the same engine, and pressure in boiler as in (7), the weight of coals burnt per minute is 12 lb., find the velocity of maximum useful effect.

Let the evaporative power of the boiler be 7·24 lb., and let the engine be condensing and belong to Class II.—*Ans.* 422 feet per minute.

(9.) When the clearance is $\cdot 05$ of the space passed through by the piston in one stroke, that is, when $c = \cdot 05$.

The area of the piston 2560 square inches, pressure in boiler 58·8 lb. per square inch. When steam is cut off at $\frac{1}{3}$rd of the stroke, and 120 lb. of steam are evaporated per minute, what is the velocity of maximum useful effect?

$$\text{Here } r^1 = 3, \text{ hence } r = \frac{3 + 3 \times \cdot 05}{1 + 3 \times \cdot 05} = 2 \cdot 74$$

$$\text{Now } \frac{144 \times 2 \cdot 74 \times 120}{2560 \times \dfrac{\cdot 264}{467}} = \text{the answer in feet per minute.}$$

(10.) Clearance $\cdot 042$, $a = 2260$ square inches, pressure in boiler $= 44$ lb., cut-off at $\frac{1}{8}$th of the stroke, evaporation 25 lb. per minute: find velocity of maximum useful effect.

(11.) Given the velocity of the piston to be 380 feet per minute, $a = 3140$ square inches, $r = 7$, evaporation $= 18$ lb. per minute: find H.P. Condensing engine of Class I. Back-pressure $= 2 \cdot 5$ lb.

$$D = \frac{144 \times 7 \times 18}{380 \times 3140} = \cdot 0152$$

From table in Art. 32, p_1 must be 5·5 lb.

$$\text{Now, H.P.} = \frac{3140 \times 380\,(5 \cdot 5 \times \cdot 423 - 2 \cdot 5)}{33000} \text{ the Ans.}$$

(12.) Find the indicated H.P. when 20 lb. of coal are burnt per minute. The velocity of the piston $= 600$ feet per minute, $a = 2400$ square inches, $r^1 = 6$, back-pressure $= 3$ lb., clearance $c = \cdot 04$ lb. Engine of Class II.

Here

$$r = \frac{6 + \cdot 24}{1 + \cdot 24} = 5 \cdot 03$$

Now,

$$600 \times 2400 = \frac{144 \times 5 \cdot 03 \times 5,400,000 \times 20}{\text{H}}$$

and

$$\text{H} = 1920 p_1 + 4000$$

hence

$$600 \times 2400 = \frac{144 \times 5 \cdot 03 \times 5,400,000 \times 20}{1920 p_1 + 4000}$$

from which $p_1 = 26 \cdot 2$ lb. per square inch.

$$\text{H.P.} = \frac{2400 \times 600 \ (26 \cdot 2 \times \cdot 49 - 3)}{\cdot 00611 \ (50324)} = \frac{2400 \times 100 \times 9 \cdot 838}{\cdot 00611 \times 50324}$$

$$= 4606 \text{ Ans.}$$

(13.) Find the initial pressure, when $a = 2200$ square inches, $r = 3$, $w = 30$ lb., $v = 320$ feet per minute. Engine of Class II. . Evaporative power of boiler, as usual, $7 \cdot 24$.

$$\frac{4752000 \times 30 \times 3}{\cdot 00611 \times 320} = 1920 p_1 + 4000$$

from which we get the value of p_1.

(14.) Find the initial pressure, when $a = 3400$, $r = 5$ $\text{F}^1 = 206$, $v = 560$. Engine of Class III., II., or I.

$$206 = \frac{3400 \times 560 \times \text{D}}{144 \times 5} \therefore \text{D} = \cdot 078$$

from which we find the answer $p_1 = 31 \cdot 3$.

(15.) Find H.P., when $w = 26$, $v = 325$, $a = 1850$, $r = 4$. Back-pressure $p_3 = 1\cdot6$. Condensing engine of Class I. (*See* Example 12 above.)

(16.) Find E^1, the evaporation in pounds per minute, when H.P. $= 314$, $V = 260$, $a = 1720$, $r' = 7$, clearance $= \cdot04$, back-pressure $p_3 = 1\cdot7$. Evidently $r = 5\cdot7$. Hence, $M = \cdot468$. Engine of Class III.

$$p_1 = \frac{33000 \times 314 + 1720 \times 260 \times 1\cdot7}{1720 \times 260 \times \cdot468}.$$

Hence D may be found, and therefore E^1, or the evaporation in pounds per minute.

(17.) Find W, given H.P. $= 2160$, $V = 1960$, $r^1 = 7$, $p_e = 1\cdot6$, $C = \cdot033$. Engine of Class II.

Evidently $r = 5\cdot87 \therefore M = \cdot446$

$$450 \times 960 = \frac{144 \times 5\cdot87 \times 5400000 \times W}{1920\, p_1 + 400}$$

Again

$$2160 = \frac{144 \times 5\cdot87 \times W\,(p_1 \times \cdot446 - 1\cdot6)}{\cdot00611 \times 1920\, p_1 + 400}$$

From which two equations, on eliminating p_1, we get the value of W.

BOOK III.

LOCOMOTIVES.

BOOK III.

LOCOMOTIVES.

CHAPTER I.

HISTORY OF THE LOCOMOTIVE.

181. RAILWAYS were in use about coal-pits to facilitate the carriage of coal a long time ago. It was not till a century ago that the rails, which had been of wood, began to be shod with cast iron. These iron rails were at first merely protectors of the wood, but in a short time they became of greater importance, were made heavier, and their material was everywhere of wrought iron.

182. WATT, in 1759, suggested the application of the steam-engine to traction on railways. He designed a condensing locomotive, and actually proposed the introduction of tubes.* But his time was too much occupied in improving the condensing-engine to allow of his actually constructing a locomotive. He knew that the great weight of a condensing-engine of sufficient power would unfit it for this purpose, and he had little inclination to think of *any* application of non-condensing-engines.

* Colburn's Hist. of Loc.

Watt's friend, MURDOCH, constructed a little high-pressure engine, whose fly-wheel became the driving-wheel of a carriage, which was made to run rapidly along a road. The driving-wheels were 9½ inches in diameter.

M. CUGNOT in 1769 constructed a locomotive. It was very defective.

TREVITHICK, in 1802, patented a high-pressure locomotive. Here the crank-shaft had a fly-wheel, and worked the driving-wheels by means of intermediate gearing. He introduced a pair of bellows for the fire, and worked them from the crank-shaft. With his light engine he found that he could not depend on the adhesion of the driving-wheels to the rails, and he suggested the use of teeth.

BLENKINSOP, working out the ideas of Trevithick, constructed an engine which had a maximum speed of 10 miles per hour.

CHAPMAN, in 1812, used a chain instead of the toothed rail. This chain was fastened at the two ends of the line of rails, and was coiled round the driving-wheel of the locomotive.

BLACKETT, in 1812, found that proper distribution of the weight enabled sufficient friction for tractional purposes to exist between the driving-wheels and smooth rails, and from this time forward engineers confined their attention to methods of driving the wheels.

It was recognized that, for convenience in stopping and starting, locomotives ought to be provided with two cylinders, their cranks being at all times at right angles to one another.

STEPHENSON constructed the *Killingworth Engine* in 1814. This was a modification of Blenkinsop's geared engine. The two crank-shafts of a double-cylinder engine were connected with two driving-wheel shafts, and with one another by means of five spur-

wheels, which also kept the cranks at right angles to one another.

To remedy the great defects of noise and wear, Stephenson dispensed with the spur-gearing, putting cranks on the driving-wheel shafts, and, to preserve the relative positions of the cranks, he used an endless chain, working over pulleys which were cogged to catch the links.

After a time, outside coupling-rods, connecting pins on the driving-wheels, were employed to preserve the relative positions of the cranks.

183. From 1814 to 1829 improvements were constantly being made on the old Killingworth engine, with its long connecting-rods, and its upright cylinders partially immersed in the boiler.

The slow speeds at which these engines were driven rendered a quick draught in the furnace unnecessary; and although Trevithick employed a pair of bellows, and then *a jet of exhaust-steam* in the chimney, and though Chapman and others had *fans*, the use of these means for obtaining an artificial draught in the furnace was almost discontinued at this time. It may, however, be mentioned that Hackworth, in 1827, made an efficient application of the blast-pipe to his chimney.

The principal changes effected in the Killingworth engine till 1829 were the introducion of steel bearing-springs over the axle-boxes, of wrought-iron tyres to the driving-wheels; and of the bogie system, in which the great weight of the engine was borne by two carriages connected with each other by swivels.

On the opening of the Stockton Railway for the conveyance of passengers, engineers were made to understand that lightness and strength had now become of the greatest necessity.

Hackworth, the engineer of this line, contests with Trevithick the honour of having made most improvements on the locomotive before 1830.

184. Two of Stephenson's locomotives had been sent to France. They were fitted to run at the mean rate of four miles per hour.

M. SEGUIN, a French engineer, experimented on these engines, getting better results from them by increasing their heating surface, *using, instead of a large flue, a number of tubes* from the furnace to the chimney. M. PELLETAN suggested a steam-jet for the chimney in connection with the tubes; and in these two improvements Seguin and Pelletan did much for the locomotive.

185. Unaware of the improvements which were being effected in France, the Manchester and Liverpool Railway Company, in 1829, offered a premium to engineers for the construction of the best locomotive engine; which, on a level railroad, should draw three times its own weight at the rate of 10 miles per hour. The engine was to have six wheels if its weight exceeded $4\frac{1}{2}$ tons; the weight being in all cases carried on springs. There was to be no smoke produced, the pressure of steam not more than 50 lb. per square inch. Two safety-valves were to be provided, and the engine was to cost less than 500*l.*

The competitive locomotives were Stephenson's *Rocket*, Hackworth's *Sanspariel*, and the *Novelty* of Braithwaite and Ericson.

The *Rocket*, shown in Fig. 55, had 25 tubes of 3 inches diameter from the fire-box to the chimney, and (in imitation of the *Sanspariel*) was provided with a steam-jet for the chimney on the eve of the day of trial.

The *Rocket* had a cylindric boiler, 6 feet long and $3\frac{1}{4}$ feet in diameter; the furnace being 2 feet wide and 3 feet high, surrounded by water. Two cylinders of 8 inches diameter. Stroke, $16\frac{1}{2}$ inches. The driving-wheels were 4 feet 8 inches in diameter.

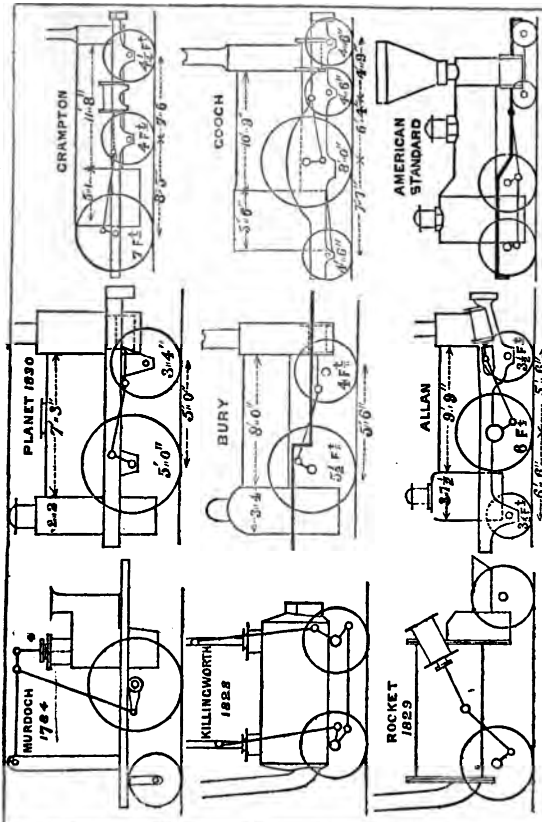

MURDOCH
1784
KILLINGWORTH
1828
ROCKET
1829
PLANET 1830
BURY
ALLAN
CRAMPTON
GOOCH
AMERICAN
STANDARD

FIG. 55.

The *Sanspariel* had a steam-jet in the chimney and one cylindric flue in the boiler.

The *Novelty* had one long tube of 3½ inches diameter traversing the length of the boiler three times. A draught was created by means of a pair of bellows placed in the chimney.

The *Rocket* was the only engine which ran the whole distance appointed by the directors. It had an average speed of 13 miles when loaded, and maximum speeds of 24 miles and 35 miles per hour when loaded and unloaded. A series of accidents prevented the principles of construction of the other engines from having a fair trial.

On these early engines began the series of reliable experiments which have been made on locomotives to determine the proper proportions of heating surface, draught, &c., required under different conditions (Art. 142).

186. Stephenson, on the Manchester and Liverpool Railway, proceeded to construct engines with more tubes of smaller size, and with larger cylinders, but still with the whole engine square and short, and the cylinders outside, as in the *Rocket*. From the great distance apart of the crank-pins, these engines had much of an oscillatory motion.

Hackworth, on the Stockton Railway, constructed engines with inside cranks and cylinders; using in the boiler, in one case, a flue which had water-pipes crossing it. In a series of engines he conveyed the smoke through the boiler from the fire-box by a large flue, and brought it back again to a chimney near the fire-box by means of tubes. This proved a very good arrangement.

Bury made a six-wheeled locomotive, but, as it was heavy and hurt the rails, he constructed a lighter engine. This had four large coupled wheels, and two

inside cylinders. It was extensively copied, and, after many improvements, it became the *Bury Engine.*

Stephenson's *Planet* (Fig. 55) of 1830 combined all the then known improvements, having inside horizontal cylinders placed in the smoke-box, and a multitubular boiler and blast-pipe.

Two driving-wheels of 5 feet; two carrying-wheels of 3 feet; boiler $6\frac{1}{2}$ feet long, 3 feet in diameter, containing 129 tubes of $1\frac{5}{8}$ inches diameter; cylinders 11 inches; stroke 16 inches; weight of engine when empty, 8 tons; when charged, 9 tons.

187. At this date begins a new era in the history of the locomotive. Engines of the *Planet* class were so heavy that the rails gave way; so that new, heavier rails were necessary, and began to be introduced. The driving-wheels, or the wheels on the crank-shaft, placed as they were before the fire-box, had most of the weight of the engine to carry, and conducted excessive stresses to the points at which they touched the rails. This was slightly remedied by keeping the front wheels close to the driving-wheels; but the engine now became top-heavy and rocked alarmingly, even at ordinary speeds. To prevent *rocking*, auxiliary wheels were placed behind the fire-box, not to help to carry weight, but to keep the engine steady. The elasticity of the framing in the *Planet* class of engines allowed the locomotive to conform perfectly to curves in the line of rails.

The introduction of wheels behind the drivers not only prevented rocking but diminished the lateral stresses on the driving-wheels, which were afterwards made without flanges. These lateral stresses had been very wearing on the boiler, and the engine was now enabled, through the elasticity of its framing, to conform perfectly to curves in a line of rails, whilst no material strains were produced in the boiler itself.

Six-wheeled engines now became common, and as the rails were gradually increased in weight from 35 to 50, and from 50 to 75 lb. per yard, the locomotives on the Liverpool and Manchester line increased in weight from 5 to 7 and from 7 to 12 tons.

In 1836 the locomotives on the Manchester and Liverpool line were very much alike, resembling the *Planet*, the average dimensions being: cylinders 12 inches diameter, stroke 17 inches, driving-wheel 5 feet, and steam 50 lb. pressure. When at the average speed of 18½ miles, the consumption of coke was 0·31 lb. per ton drawn per hour. When at the speed of 29 miles, the consumption of coke was 1 lb. per ton per hour.

It was not then thought advisable to use driving-wheels of more than 5 feet in diameter for high speeds, whereas in express-engines we now employ large wheels of 8 feet in diameter in some cases.

Some attempts at this time to substitute coal for coke failed completely.

It has been shown since then that coal and its smoke may be economically burned in locomotives, with careful stoking.

Fig. 56 will give a good idea of the general construction of the locomotives in use in 1836, and, indeed, of all ordinary engines in use till the present time. Changes in construction have occurred since then to suit special conditions of gauge, speed, load, compactness, &c., but the original form is seldom far departed from even in the latest locomotives, and is that given in the figure.

On the London and Birmingham Railway, Bury introduced his four-wheeled engines, with circular fireboxes and inside frames. They had a light appearance. The smoke-box was subjected to great strains, as it always is when an inside framing is employed; and besides this, the Bury engine had the faults common to all four-wheeled locomotives. Till 1845 only one six-wheeled engine was employed on Bury's line.

Bury believed that the fire-box ought to be convex to the engine-driver, instead of being flat; in fact, his fire-box was circular in plan. Again, he arranged his tubes on the tube plate in arcs of circles, for the sake of better circulation in the water of the boiler. These peculiarities are now known to be immaterial in effect, and when we consider the much lower price of the square fire-box, and the greater number of tubes which may be introduced in the straight-line arrangement, we see that Bury's refinements ought not to be attended to.

The relative values of inside and outside framings will be discussed in Art. 216.

Bury's arguments in favour of the use of four wheels are weak, and time has shown their uselessness (Art. 216).

These four-wheeled engines were sometimes as heavy as 13 tons; but the usual weight was 11 tons. The average dimensions were:—Cylinder 13 inches diameter, stroke 18 inches, and driving-wheels 5 to 6 feet diameter.

In 1837 CHURCH, of Birmingham, constructed a steady four-wheeled locomotive, in which the driving-wheels were placed *in front* of the others. These engines carried water-tanks and coal bunkers. Church was the first to construct a four-wheeled tank-engine with tubular boiler, and was the first to prove the advantage of large six-feet driving-wheels, and of large steam-domes.

The cylinders were outside, and were placed behind the engine under the foot-plate, instead of being in front in the smoke-box; so that the fire-box was no longer overhung, and the driving-wheels might be made to bear whatever proportion of the whole weight might be determined on; by this means one great objection to the use of only four wheels was removed.

Again, this engine *rocked* but little, for its centre of gravity was brought very low by the arrangements in position of its tanks and bunkers. Instead of having a wheel-base of 5 feet, like the ordinary four-wheeled engine, this had one of 8 feet in length. On the working engine the driving-wheels usually conveyed 9 tons, and the hind wheels 5 tons of the weight to the rails.

In 1840, Norris, of Philadelphia, constructed six-wheeled

Bogie-engines. In these the two pairs of small front wheels were framed together, and were able to turn slightly independently of the rest of the engine.

These Bogie-engines were designed for lines with steep gradients and quick curves.

188. A necessity now began to be felt for powerful engines, and these must have larger cylinders and altogether larger parts. The gauge, or inside distance between the rails, was usually 4 feet 8½ inches—a small space in which to pack two cylinders with their valves and gearing, so that several rash changes in the gauge were made to suit the convenience of constructors of locomotives.

Alteration in the gauge is of so much consequence that Locke, of the London and North-Western Railway, preferred to re-introduce outside cylinders.

The *Allan Engine* had large driving-wheels, and a compact arrangement of machinery.

The frame is inside, but passes through the smoke-box; the inclined cylinders are placed outside the smoke-box, and are attached to the frame itself. The connecting-rod works a pin in the driving-wheel, instead of working an ordinary crank-pin. As there are no internal cranks, the driving-shaft is straight and quite close to the boiler-shell, so that the wheels are of 6 or even of 6½ feet in diameter. A thin supplemental outside frame (Art. 216) is a marked feature of the engine. Locomotives similar to this are still maintained on many lines, and in the Allan we see the first of a series of powerful engines, whose construction still receives the particular attention of engineers.

The Allan engine was provided with a link-motion for expansive working.

188*a*. Contrivances for distributing steam to the cylinder have an independent history.

In Trevithick's time the eccentric and D valve were well known as parts of the ordinary non-condensing steam-engine.

A square cam was used for the fore and back working of the valves in the *Killingworth Engine*, but as this was defective, an eccentric was introduced whose position could be altered on the crank-shaft (in fact, the common moveable slotted eccentric worked by a lever fixed on the shaft).

HACKWORTH used similar loose eccentrics, bolting them together; and, with modifications, this was the usual valve-motion arrangement for a length of time.

CARMICHAEL used a fixed eccentric, the end of whose rod was fitted with a double fork, which might be made to catch and hold one pin of a certain lever for fore- and another and opposite pin for back-working; so that, in both cases the *lead* of the valve was maintained the same. Had this not been cumbrous, the four-eccentric arrangement would never have been substituted.

In 1837, two eccentrics were given to each cylinder, each rod having a fork, and for a long time engineers merely directed their attention to simplifying this valve motion.

At length *Stephenson*, removing the valve-chests from above to the space between the cylinders, was able to put the forks on the valve-rod itself, pins on the eccentric-rods being lowered or raised into their proper places by levers in the usual manner.

The importance of allowing expansion of steam in the cylinder began to be understood in 1840. And it was seen that expansion might be effected by means of an ordinary valve, provided with a proper amount of lap and a variable travel. Gray, and afterwards Cabrey, contrived complex forked-valved arrangements, giving variable travel; but the *link-motion* designed by *Williams*, and perfected by *Howe* in 1840, is that now universally employed in locomotives.

For other methods of allowing expansion of steam in the cylinder, *see* Arts. 88—97.

188*b*. GOOCH used seven-feet wheels on engines for express trains, and the Gooch locomotive is still employed on the South-Western Railway. Stephenson introduced very long boilers, but their evaporative powers did not answer his expectations; besides, from the length of the boiler, the

cylinders were bolted *behind* the smoke-box under the barrel, and this method of fixing gives rise to unsteadiness at high speeds. These engines had inside frames formed of one stout bar.

Brunel, in 1833, had arranged for a seven-feet gauge on the Great Western Railway, and thus considerably simplified the construction of his locomotives, which may be seen in the powerful eight-wheeled "Great Britain" class of engines with inside cylinders, designed by Brunel and Gooch.

We see that the great point with later locomotive engineers was to obtain great power with large wheels and low centre of gravity.

CRAMPTON made the greatest modern improvement when he brought all the machinery outside, enlarged the fire-box, and *passed the crank-shaft behind the fire-box altogether.*

Hitherto the low centre of gravity of the engine had prevented the use of large wheels, but Crampton's boiler-shell is immediately over the carrying-wheel shafts; so that, as the powerful engine is necessarily long, and has four pairs of wheels, the stability is as perfect on this narrow-gauge locomotive as in any of the broad-gauge locomotives of Gooch.

In useful engines recently constructed, the cylinders, whether outside or inside, are seldom more than 17 inches in diameter, the stroke is never more than 24 inches, and the driving-wheels are never less than 5 feet, and seldom more than 7 feet in diameter.

In six-wheeled engines for high speeds the wheels are often equal in size, and are coupled together. Where there is good workmanship and strict attention as to repairs, the permanent way being good and with few curves, this arrangement is satisfactory. With quick curves on a railway, it is best to use engines of which only two pairs of wheels are coupled.

189. **The Locomotive in America.**—Till 1840, the engines run on American railways were usually imported from England, or, if built in America, were constructed to drawings of existing English locomotives,

slight modifications being introduced to suit the special characters of the fuel and permanent way.

The house for the engine-driver, the cow-catcher, the large bell, and the spark-arrester, used in all wood-burning engines, are distinctive features of modern American locomotives.

The permanent way is weaker than in England, and has often sharp curves; hence two pairs of coupled driving-wheels are now always employed, two pairs of small bogie-wheels being placed under the smoke-box.

The lap on the valve is usually much less than in England, cut-off being seldom less than one-third of the stroke. It may be mentioned that the blast-pipe is smaller when wood is burnt, these wheels being annealed after being cast in a chill; that brake-blocks are often of cast iron, because non-conducting wood may counteract the effect of annealing the wheels; that the whistles are larger; that, instead of glass gauges, four or more gauge-cocks are employed, and that the Americans commonly use cast-iron wheels for the sake of cheapness.

CHAPTER II.

GENERAL DESCRIPTION OF THE LOCOMOTIVE.

190. For a general description, nothing is better suited than the skeleton drawing (Fig. 56) of one of Stephenson's six-wheeled engines; first, because the construction there adopted is seldom departed from, even in the most modern locomotives; and, secondly, because the valve-chests are placed above the cylinder; and hence, the connections between different parts may be easily recognized.

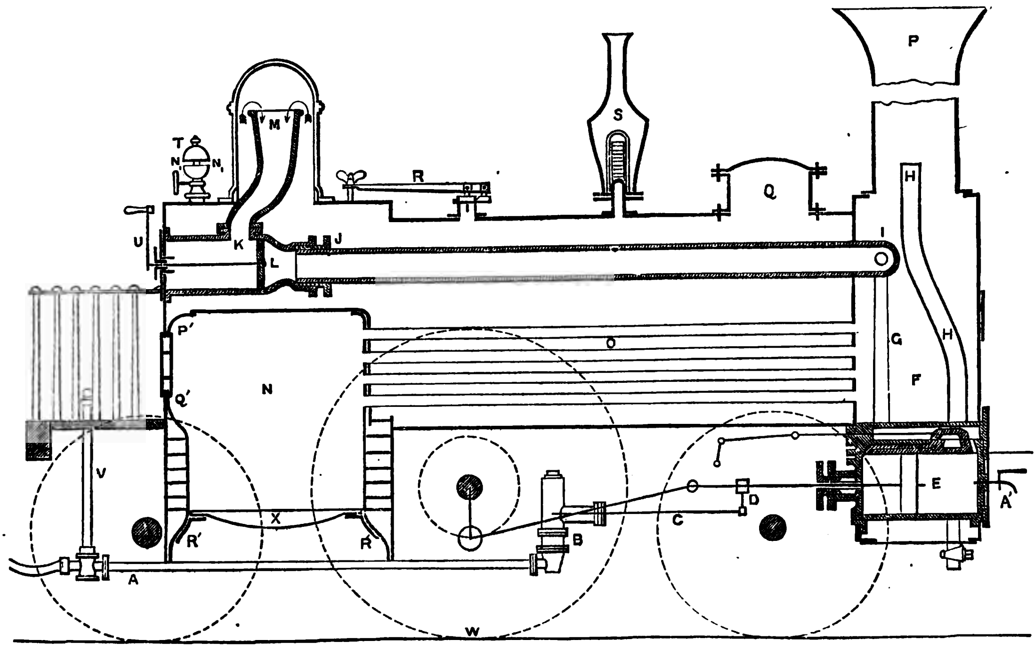


FIG. 56.

The boiler consists of—N the fire-box, O the barrel, and G the smoke-box. Fuel, which is coke or coal, or a mixture of both, is laid on the fire-box X through the double door P′ Q′. The fire-box is surrounded on nearly all sides by water-spaces. The hot gases go along the tubes O, of which there are a great many, giving off their heat in passing through the barrel of the boiler on their way to G the smoke-box, from which they escape by the chimney P. The draught is helped, or almost wholly created, by the escape of exhaust-steam into the chimney from the blast-pipe H.

The man-hole door Q, the safety-valves S and R, the whistle N, and the dome over M, are other appendages to the boiler.

Since the steam-space is small, it is necessary, to prevent priming, to take steam from the top of the dome. It passes by the regulator L through L I G, to the valve-casing of the cylinder E.

The slide-valve distributes the steam, which leaves the cylinder by the exhaust-port and escapes up the blast-pipe H to the atmosphere. The piston E gives motion to a cross-head and a slide, and so to the connecting-rod and crank, thus turning the driving-wheel W. The other pairs of wheels serve to support the engine. The axles of the wheels work in bearings attached by springs to a frame, which keeps all parts of the engine rigidly in their proper positions, and corresponds to the foundations and pillars and beams of ordinary stationary engines.

The force-pump plunger C is attached to the piston-rod at D, and has the same stroke as the piston. Water is forced into the boiler by means of this force-pump, and comes from the tender, or attendant carriage, through the pipe A.

Roof-stays strengthen the top of the fire-box.

The water-space surrounding the fire-box is called the *fire-box shell.*

The *heating cock* V lets feed-water come from the tender.

The *blow-off cock* is in the fire-box shell to let the water escape from the boiler.

The *ash-pan* is beneath R.

The *transverse stay-plate* binds the two sides of the frame together, and supports brackets, guide-bars, &c.

CHAPTER III.

DETAILS.*

191. **Cylinders and Valve-Chests.**—The cylinders are made of either good cast iron or of cast iron toughened with wrought iron, as in Gooch's cylinders. They are provided with long flanges for attachment to the framing, and in outside cylinder-engines these attachments are made in many different ways, in which the frame is considerably stiffened.

The valve-chest is usually vertical, and is cast in a piece with the cylinder; and in inside cylinder arrangements there is much compactness when the two valve-chests are bolted together, and the flanges, whether at the sides or above and below, bolted to the framing.

Allan has long double flanges, by means of which his outside cylinders are bolted to a double frame fitted at that place with transverse stay-plates.

The hinder end of the cylinder, or that towards the crank, is usually cast with an inside circular flange, leaving an opening smaller than the piston; this open-

* Readers will benefit greatly by examining the Plates in Mr. Clarke's work on Locomotives.

ing being closed from the outside with a small cylinder-cover, provided with the stuffing-box. This arrangement makes the cylinder very strong. The strength is often increased by the front being cast in one piece with the rest of the cylinder. On the continent it is common to pass the piston-rod through the front of the cylinder for steadiness.

There must be as little clearance as is possible, and both steam- and exhaust-passages must be wide and direct. That the piston may not overhang the edges of the openings, the passages are cut into the ends of the cylinder, as shown in Fig. 15.

And yet the clearance ought never to be much less than $\frac{3}{8}$ of an inch in any engine ; priming being liable to occur, and the steam often condensing.

It is to remove the condensed water that small water-cocks open from the lower parts of the cylinder and sometimes of the valve-chest. These *water-cocks* are opened by means of a handle at the foot-plate. The condensed steam in the cylinder lubricates the piston, and on hilly lines where the engine may be required to work while there is no steam in the cylinder, unless oil has been injected through the water-cocks, or unless there is a grease-cock, the piston may heat up very greatly indeed.

Mr. Widmark, of Bristol, has patented a good arrangement of water-cocks, in which a plug is moved by steam-pressure, and opens the cock at the command of the engineer. A small steam-pipe communicating with the different cocks on the boiler opens them all at once when a cock in front of the boiler is turned.

The valve-face table is raised above the surrounding level, and it is necessary that the valve should overshoot this table at every stroke, for the purpose of promoting uniform wear.

Sometimes an intermediate steam-chest is bolted between the two cylinders. The chest-cover is at one end or the other, or else below, as there is little or much room.

In American locomotives, the cylinders are usually cast open at both ends, the covers being large, with their flanges outside. The steam-chests are always cast separately, and bolted to the cylinders.

192. **Piston, &c.**—The body and cover of the piston are usually made of cast iron or brass, the latter being considered most suitable, as it is not so apt to break when sudden resistance is offered to its motion. It is well that the piston should break rather than the cylinder at a sudden check, and hence wrought iron is considered too strong and cast iron too weak for the body of the piston.

Cast iron is the best material for rings, as it is not so liable to fracture as steel, and is more durable than brass, which is only used for the rings in cylinders made wholly of wrought iron or of soft cast iron.

Steel cuts the cylinders, whereas cast iron keeps them polished. The metal used in rings by Allan has $2\frac{1}{2}$ oz. tin and $\frac{1}{2}$ oz. zinc per pound of copper.

As the cylinder lies on its side, it is likely to wear oval in time; and to accommodate the piston to this change, the rings are often divided into four pieces. The form of ring most commonly adopted is that shown in Fig. 28, where we have a single-cut ring, thick near the cut, and sprung with a wedge and continuous steel spring.

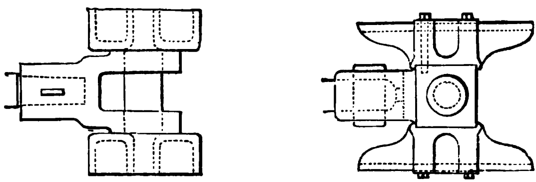
The *cover*, or what corresponds to the *junc-ring*, is bolted to the body of the piston with screws that cannot shake loose, as they have little sheet-iron covers. Fairbairn inserts brass nuts in the body to receive these screws.

Piston-rods should be made of Low-Moor wrought iron, as this is less liable to snap than steel; and it is said that the best pistons are forged in a piece with the piston-rod. The methods of fixing described in Art. 102 are often employed; for others consult the drawings in "Clark's Locomotives."

In America the body of the piston is usually of cast iron; the rings have single oblique cuts without wedges, are made of brass (whose composition is 9 of copper to 1 of tin), and are pressed out with several small springs, like the piston of Fig. 27.

193. **Cross-heads, &c.**—Cross-heads are of wrought iron; they are provided with slides which work on the guide-bars; they have either one or two journals for the end of the connecting-rod, and a socket for the piston-rod. There may be two slides and two pairs of guide-bars (Fig. 57), or one slide with one wide vertically-placed pair of guide-bars (Fig. 58).

The former is considered best, as, although it only admits of a short connecting-rod, this is single-ended.



FIGS. 57 and 58.

The piston-rod is usually tapered and keyed into the cross-head (Figs. 57 and 58), although in some cases a nut is used.

The *slides* are commonly of cast iron, and as they are merely bored to receive the cross-head pins, may readily be removed.

The *guide-bars* are bolted to brackets on the hind-cylinder cover, and to brackets on the transverse stay-plate.

If the slides are to be faced it is best to use brass, for steel and case-hardened wrought iron are liable to cut up the guide-bars. Guide-bars are $2\frac{1}{4}$ to 3 inches broad if placed

horizontal, or 4 to 5 inches if vertically placed. Steel and case-hardened wrought iron are most common, but many good guide-bars are to be found made of cast iron.

In America, where fewer brass slides are used, the bars are always case-hardened or of steel. The cross-heads are sometimes of cast iron, with a pin cast in, the slides forming part of the casting.

194. **Connecting-rod.**—The rod should never be less than five times the length of the crank, and is rarely less than six times as long. In America it is never less than seven times its length. In section it varies with different makers, from flat to being almost circular.

It is always of good wrought iron, with brass bearings, provided with steel keys. Fig. 33 shows a common form. The crank-end is much larger than the other, and its strap R is often in one piece with the butt. Fig. 36 shows the head of an outside connecting-rod where the strap is in one piece with the butt.

This form with the fixed strap is to be recommended, being stronger and allowing of more easy cottering up, which may be done at the small end; yet many makers still adhere to the movable strap.

The style of construction of the small end is comparatively unimportant, but it ought to be simple and have few parts.

Outside connecting-rods for two pairs of driving-wheels should have their heads of the shape shown in Fig. 36.

To connect three pairs of wheels of the same size, separate rods are provided for each pair, and not one continuous rod for all three.

Oil-cups are forged on the heads, and are fitted with syphon-tubes and brass covers. The swing of the head will usually spill enough oil into the tube, but some makers use a capillary thread.

CHAPTER IV.

DETAILS (*continued*).

195. **The Slide-valve.**—The three-ported valve, or "locomotive-slide," gives remarkably satisfactory work when furnished with the link-motion, for we are enabled to employ accurately almost any grade of expansion, and to change rapidly from one grade to another, the parts being well bound together.

Slide-valves are usually of brass, but often of cast iron, which is in many ways better, but is liable to wear the valve-face.

The *valve-rod*, which is forged in one piece, is attached to the valve by a large surface, and is often for steadiness continued through the front of the valve-chest. A spring keeps the valve pressed down on its table. The eccentrics are cast in halves, and keyed on the shaft, and have straps of brass or wrought iron.

196. **Link Motion.**—With good expansion gear, the pressure ought to be the same during the whole time of admission, and as nearly as possible equal to the pressure in the boiler; the valve ought to close rapidly, that there may be a minimum of wire-drawing; the pressure at the end of the stroke and during the back-stroke ought to be nearly equal to that of the atmosphere.

In locomotives the number of strokes per minute, and the pressure of steam employed, are so much greater than in ordinary high-pressure engines that these results are never completely attained.

The smaller the lead of the valve the less apparent is the fall in pressure through wire-drawing during the admission.

Wide steam-passages and increase in lap within certain limits decrease wire-drawing.

With higher speeds the exhaust-port is opened much sooner before the end of the stroke, that the back-pressure may be very little greater than that of the atmosphere.

From Mr. Clark's experiments, we find that when the link-motion is well arranged, there is little wire-drawing with ordinary expansion at the speed of 600 feet per minute of the piston, but that as the time of admission gets shorter the wire-drawing becomes greater and occurs at lower speeds.

There is a certain amount of opening of the steam-port, which is sufficient to keep the pressure constant at a certain speed; this is shown by the pressure being quite constant at the beginning of the stroke, when the piston is moving slowly and the port has just been opened. Hence, the smaller the lead of the valve, the less apparent is the fall of pressure through wire-drawing. Since the lead ought to be less when there is more expansion, the link-motion to be described presently is not so effective for this reason as the *stationary link*, although it has many important advantages.

When dry steam is used there is less friction in the passages, and hence the initial pressure is greater, and the wire-drawing seems increased; but in reality, with steam containing water the wire-drawing is often very considerable during the whole time of admission.

Again, with dry steam the back-pressure is inconsiderable, even at the highest speeds and with moderate periods of release. In general, the steam contains much condensed water, and so is retarded by friction at bends in the exhaust-passages. Steam is usually much drier in inside than in outside cylinder engines; but in all ordinary cylinders it is found that the back-pressure varies as the pressure at release and as the square of the speed, unless when the steam contains much water, and then the back-pressure varies with the pressure at release and the speed merely.

With inside cylinders, the back-pressure due to a given pressure at the time of release is less the sooner the release is made, and it will be remembered that the forward pressure is

not very injuriously diminished by a release some time before the end of the stroke; however, in some unprotected outside cylinders, increased time of release allows so much expansion that much steam is condensed, and by increased friction in the passages renders the back-pressure as great or even greater than before.

In engines at speeds of from twenty to forty miles per hour, with steam cut off at half stroke, the mean back-pressure is ·16 of the pressure at release for inside cylinders, and ·28 for outside cylinders partially protected.

Link motions have either the link suspended at its middle point with the block moveable and connected by a joint with the slide-valve, or the link moveable and the block immoveable. Either of these, when properly constructed, will give the same lead for both forward and backward motion; but the latter link, which is most common, is very defective in giving different amounts of lead for different grades of expansion, the lead getting greater during the change from full-gear to mid-gear; that is, as there is greater expansion allowed; lengthening the eccentric rods makes the lead less in this case.

The stationary link gives the same lead for all grades of expansion, and by properly adjusting the method of suspension may be made to suit any given conditions of expansion. In the Allan engine a link is adopted which is itself moveable, the block also being moveable. For a detailed description of these link-motions students are referred to larger treatises.*

From want of room and for other reasons the moveable link is not generally adopted in locomotives.

197. The ends of the eccentric-rods are attached to the *link* at B and C (Fig. 59), and the valve-rod B terminates in a block, which slides in the slot as the link is lowered or raised by means of the *reversing link* attached to L.

That the end of the eccentric-rod may coincide with the centre of the block in full-gear, it is common to use

* *Zeuner on Valve-gearing* (translation) is a book which will repay earnest study.

a *box-link* instead of the open link of the figure. This surrounds the block, partly hiding it from view.

Let the crank be moving against the direction of the hands of a watch, then E is the fore-eccentric and F is the back-eccentric.

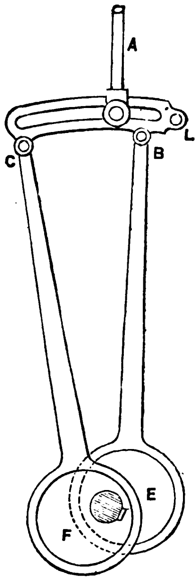


FIG. 59.

When the link is lifted, B is moved in a circle, with the centre of the disc E as centre, and C is moved in a circle, with the centre of the disc F as centre; and hence we may, without much inaccuracy, suppose that every point in the link moves in a circle about the centre of the crank-shaft. However, there is as much inaccuracy as alters the *lead* of the valve when the link is moved.

Making the skeleton drawing of this link-motion, we find that when in full gear, whether fore- or back-stroke, there is the same amount of lead, and that this is less than the lead at any other time. In mid-gear there is most lead, and for both fore- and back-strokes of the valve it is increased as the travel of the valve diminishes. The longer the eccentric-rods, and the shorter the link itself, the less is the aberration in the lead.

It is well to represent eccentrics by little cranks in skeleton drawings; the angle between these cranks will be the supplement of twice the angle of advance, as may be seen from a sketch of the positions at the beginning of the stroke. In this link-

motion it is only in full-gear, that is, when the block is at B or C, that the linear *advance* of the eccentric is equal to the linear *advance* of the valve. In all other positions of the block, since the advance = lead + lap, as the lead varies, the advance of the valve will also vary. It is best to so set the eccentrics that the valves may have the usually desired lead when in half-gear, the position shown in the figure. This is a common working arrangement; but it gives in full gear a lead which is slightly less than might be desired.

When the link is lowered till B is opposite the block, the valve gets the full stroke of the eccentric E, while F merely pushes C backwards and forwards without much affecting the motion of the valve. Again, when the link is lifted until C is opposite the block, the valve gets the full stroke of the eccentric F, while E merely pushes B backwards and forwards.

Now, as the eccentrics are placed in almost opposite positions on the crank-shaft, B is always being pushed forward as C is pushed backward: the effect of lifting the link will be to alter the motion of the valve.

If the block is midway between B and C, it will get a motion compounded of the motions of B and C, and in fact it will be found that the valve has its shortest stroke when the link is in *mid-gear*.

It is not necessary to suppose that the middle point between B and C is motionless. In fact, this could only happen if the link suspending L were infinitely long and there were no lead, so that the eccentrics were exactly opposite. The paths of the block in different positions of the link may easily be discovered by means of a skeleton drawing. When such a drawing is made, it is found that the method of suspending the link has a very great effect on the paths of the points at the different parts. Having the point of suspension L above (Fig. 60), and suspending from a shaft placed below, or again having L below and the shaft above, tend to equalize the times during which steam is admitted in the fore- and back-strokes, and are both better arrangements than that shown in the

figure. The **box-link** is more favourable than the open-link to this equalization, unless in the case in which the valve is worked by a lever from the block, and hence has a reverse motion. When the block communicates motion to the valve by means of a lever, and in its motion is just opposite to that of the valve, the eccentrics must be placed diametrically opposite to their present positions on the crank-shaft, and we have to change the terms back-stroke and front-stroke for front- and back-stroke respectively. The use of the lever produces very little alteration in the nature of the motion of the valve.

198. The middle point, or the position of the block in the link when in mid-gear, ought always to vibrate

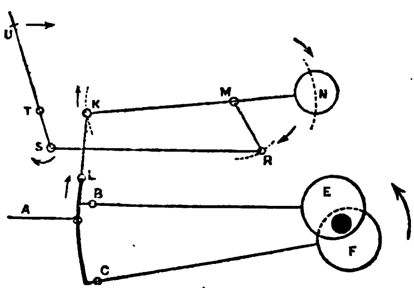


FIG. 60

horizontally, and after this assumption we may easily determine the length of the reversing lever K M (Fig. 60).

199. According to the methods of suspending the link, the fractions of the forward piston stroke, during which steam is admitted in full-gear, half-gear, and mid-gear, in an ordinary locomotive, are from ·72 to ·78, ·44 to ·48, and ·16 to ·22 respectively.

So much expansion is never allowed by the stationary link as to have cut-off at one-sixth of the stroke. But as the *release* with the moveable link takes place at half-stroke, the steam really expands to only *three times* its volume.

When more than ·75 of admission is required, the block may be shifted beyond B and C to the extremities of the link.

200. Fig. 60 shows the command which the engineer has over the position of the link. By means of the handle U, he moves S, and by the link R S, he moves the piece K M N R in the way shown by the arrows.

This piece is called the reversing lever, because by means of the *reversing* link K L, the link L C may be lifted, and the motion of the piston reversed.

The counter-weight N nearly balances the weights of the link and eccentric rods hanging from K.

The handle U T moves over a notched sector, every notch corresponding to a different rate of expansion. These notches must be closer together in the middle if we would have the same difference between every two rates of expansion.

In general, the valve-gearing is of wrought iron. The eccentric rods are usually $\frac{3}{4}$ inch thick, and are welded to wrought-iron straps or rivetted to brass straps, unless we use bolts, which render a screwed adjustable joint in the middle of the rod unnecessary.

The link is commonly in one piece 2 inches thick, the slot being 2 inches broad. The centres of the ends of the rods are $2\frac{1}{2}$ to 3 times the throw of the valve apart. The block is of wrought iron. Sometimes guides are forged on the supporting links to prevent the two sets of valve-motions touching.

In America it is usual for the links to have the shape shown in Fig. 59. Sometimes a solid bar grasped by a sliding-box on the end of the valve-rod is used. Instead of using a counter-weight many makers use a suitably placed steel spring.

Some American engineers still employ separate expansion valves, such as were described in Art. 96; but it is not probable that the benefits arising from accurate distribution of the steam compensate for the increased noise, friction, and wear.

CHAPTER V.

DETAILS (*continued*).

201. **Steam-Pipes.**—The steam-pipe L I (Fig. 56) is of copper, $\frac{1}{8}$ in. or $\frac{3}{16}$ in. thick, with brass flanges soldered on wherever flanges are necessary. The parts from M and K to J are of cast iron.

The pipe splits into two parts at I, each of which leads to a valve-chest. When the cylinders have one common valve-chest, of course there is no division of the steam-pipe.

The *regulator* is of brass, and is variously made. It is usually a flat plate which has apertures, and this plate slides over ports. It is placed in different parts of the boiler by different engine-makers. The regulator is moved by a handle working steam-tight through a stuffing-box, as shown at U (Fig. 56).

The pressure of steam in the cylinder is never equal to the pressure in the boiler, in consequence of the number of bends in the steam-pipes.

The diameter of the steam-pipe ought to be the same everywhere, viz., one-third to one-fourth of the diameter of the piston. The regulator is sometimes a flat valve sliding over ports, and is applied either in the dome or in the smoke-box. In Allan's engine the valve is conical, fitting a conical seat, and is moved by a peculiar spring. It seems to work very well.

202. **Feed-Pumps.**—These are either (1) in inside-cylinder engines, *long-stroke* pumps, worked from the cross-head and placed outside (and hence liable to be burst in frosty weather) or (2) in outside-cylinder engines, *short-stroke* pumps, worked from the back eccentric, or from a special eccentric on the crank-shaft.

Because of the great working stresses, pumps ought to be bolted to the frame and never to the boiler.

The feed-water should enter the boiler near the smoke-box: first, because cold water will have less cooling effect in contractions at the place which is coolest already; secondly, because the boiler is fixed firmly to the frame at the front, and hence its expansion and contraction will in this case produce no stresses in the delivery-pipe.

In America the delivery-pipe is almost always a flexible hose. In England ordinary metal sliding-pipes (Fig. 61) with ball-joints are used.

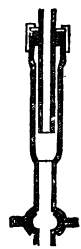


FIG. 61.

Fig. 62 will show the construction of a long-stroke force-pump for a locomotive.

The feed-water comes in at A, and lifts the ball B, when the plunger D is withdrawn, filling the space above B. The guard C prevents the ball from rising too high from its seat. On D being pushed in again, the water lifts E, and flows through F to the boiler. There is another ball-and-seat valve in the delivery-pipe F, near the boiler, to prevent the leakage of steam to the pump.

Conical valves, with guides, may be substituted for the ball-and-seat valves shown in the figure.

The valve ought to have independent covers for inspection. The pump itself is usually of toughened cast iron, or of brass; the plunger being made of wrought iron for the long stroke; and of brass for the short stroke.

with the steam through another conical mouthpiece I, crosses the open space from I to J L, which it enters, filling the expanding part L, and entering the boiler through the valve W. K is the overflow-pipe.

By means of handles the cones D and G may be brought closer together, and closer to the cone I, thus regulating the supply of steam and water.

The steam is partially condensed, and hence there is a loss of pressure to the boiler, since the volume of water entering the boiler is less than that of the issuing steam; and, in fact, the work of supplying the feed-water is part of the energy lost by the steam.

To find the smallest area in square inches of the conical mouthpieces, divide the cubic feet of feed-water per hour by 800 times the square root of the pressure of the steam in atmospheres.

The expenditure of steam at ordinary pressures in working the injector is about 14 times the volume of water injected. From these data students may calculate according to the methods of Book I. the temperature of the feed-water leaving the injector.

The injector has been much simplified lately in principle, particularly in its application to locomotives. That constructed by Mr. Webb, of Crewe, has much merit of this kind.

CHAPTER VI.

DETAILS (*continued*).

204. **The Boiler.**—The boiler is firmly bolted to the frame in a way that allows of its expansion under heat.

The connections are at three places on each side:

at the fire-box, mid-barrel, and smoke-box, and consist of rivetted bars of iron forming strong brackets.

The *shell* of the boiler is formed of plates of good wrought iron, rivetted together with single rows of rivets. Angle irons usually connect the plates at the corners, though the plates are very often rolled at the corners into flanges as a safer arrangement.*

*Boiler-plates are $\frac{3}{8}$ or $\frac{1}{2}$ inch thick, and are of a moderate width, breaking joint along the boiler. Joints are either formed by rivetted plates, lapping in the ordinary way by their edges being planed and butted against each other, and the seam covered by a strap or welt, or by the plates being welded together by a scarf or lap. Welts, or straps, seem at present to be stronger than other rivetted joints, most likely from needing fewer rivet-holes; but it seems likely from experiments that scarf-welded joints will eventually prove best.

The dome is worked out of one sheet, and is furnished with a dished cover of cast or wrought iron bolted on. For the Crewe steel boiler the dome-cover is beaten out of a steel plate.

205. The *man-hole* has one cylindric flanged plate of wrought iron thicker than the rest of the boiler, and planed to receive a wrought-iron or cast-iron top.

The *fire-box* is of copper plates in England, and wherever coal is burnt, or flaws in iron become greater under the intense heat.

The *tubes* are best when made of brass or copper, but iron is sometimes used. The end of the tube fits a hole in the tube-plate, and when in its place the end is widened by a mandril till it overlaps the sides of the hole, and a ferule of steel or iron is driven into the tube to keep it tight. Ferules may be dispensed with at the smoke-box end; they prevent cinders getting through.

* This is copied from Book II.

Tubes wear most at the fire-box end. There ought to be some tubes as high as the top of the fire-box, so that if the crown gets uncovered with water, the tubes will melt sooner.

A *fusible plug* of lead is often screwed into the crown of the fire-box, so that if it gets uncovered it shall smelt and give warning; but practically its melting cannot be depended upon.

Iron is suitable enough for the fire-box of a wood-burning locomotive. Iron blisters, partly from its want of conducting power, and it may be seen that thin plates do not blister so readily as thick. The tube-plate is usually $\frac{3}{4}''$ or $\frac{7}{8}''$ thick. A form of fire-box easily stayed now being adopted in many places is one in which the inside shell is flat only at the middle part of the top. The outside shell is flat for about two-thirds of its length. Vertical stays bind the shells together. For tubes copper is perhaps best, as it is a good conductor of heat and is very little affected by chemical action of impure water outside.

The fire-box plates, forming the sides of the water-space, ought to be so sloped that the water and steam may readily rise. An iron bar 2″ thick is often placed between the fire-box shell at the bottom and the boiler shell, the long rivets passing through. Sometimes an angle iron is used to join the plates; but in this case the rivet-heads are inside, and when they break are very difficult to repair. The best arrangement is that of Beattie, who used a broad wrought-iron hoop with outside flanges fitting between the plates.

The doorway is formed of a ring of brass or of wrought iron, faced on both sides for the fire-box and outer shell, and fixed with rivets.

The fire-door is double, and is better without perforations. It is conveniently placed when on a level with the top of the fuel, that is, with its lower part about 18″ to 24″ from grate.

Mr. Webb, of Crewe, forms the front, back, and sides of his steel boiler of one plate. The plate forming the top is flanged down on three sides. The tube-plate is separate, is much smaller than the barrel, and may be replaced without

disturbing the other parts. A little crane fitted to the dome enables the cover to be lifted.

206. **Inside Stays.**—The fire-box is flat-sided, and since the strength of the copper is practically small, a great many stays are provided. Numerous copper (or iron) bolts brace the fire-box and its shell together.

The crown is stiffened by strong wrought-iron ribs 4½ inches apart, standing up a little from the crown-plate, but connected with it by numerous bolts. These ribs are attached to the side-plates of the fire-box, and also to the fire-box shell, so as to take the strain from the upper corners, and to more securely bind the fire-box and shell.

The *fire-box shell* is cylindrical at top, and all the flat parts are bound together with tie bars by means of angle irons rivetted on the plates.

The tubes are supposed to provide sufficient binding for the barrel of the boiler.

207. **The Grate.**—The grate has wrought-iron bars at the height of a few inches above the bottom of the water-space. At some distance below the bars is the *ash-pan*, fitting the grate so tightly that air cannot get in at the sides. This is used for another purpose than merely collecting ashes, for in it a door is provided, the *damper*, and this allows us to regulate the air-supply of the furnace. Another *damper* in the shape of a swivelling iron disc should be provided for the chimneys of all locomotive boilers, and when we have command over the two dampers, we are enabled to regulate draught and air-supply in the completest manner. (*See* Art. 142, Book II.)

The smoke-box is of ¼-inch iron plate. It has a large door in front to give access to the tubes, and this is dished outwards slightly.

A small door let in at the top of the smoke-box is another efficient damper by which the engineer admits

air and deadens the draught. Of course doors like this must be worked by levers from the foot-plate.

208. **The Blast-Pipe.**—Since it is necessary to have quick combustion with a low chimney, the *blast-pipe* is employed. The copper pipe H (Fig. 56) is placed in the axis of the chimney, so that each puff of escape steam, expanding as it leaves H, fills the chimney like a piston, and, rushing upwards, tends to exhaust the smoke-box. By this means a current is induced by way of the ash-pan, through the fuel in the fire-box, which carries the products of combustion through the tubes to the smoke-box.

The body of the blast-pipe ought to be wider than the orifice, we find from Clark's experiments, and H ought to be below the bottom of the chimney, or else the chimney should be widened at the bottom.

To measure the pressure in the partial vacuum produced by the blast, a syphon mercury gauge is often employed, and with this the development of the pressure of the blast in pulsations may be noticed.

The loss of pressure or the *exhaust* in the smoke-box varies with the pressure of the blast, the relation being unaltered by change in cut-off in the cylinder. Again, the rate of evaporation varies nearly as the square root of the loss of exhaust in the smoke-box.

The pressure of the blast, as measured at the opening in the chimney, increases, it is found, with the exhaust pressure in the cylinder at the point of release, and hence (*see* Art. 196) varies with the square of the speed. It is, however, very small compared with the pressure at the point of release, or the back-pressure in the cylinder.

Within certain limits, diminution of orifice increases the draught, but as it also considerably affects the

back-pressure, the efficiency of the engine may suffer.

In practice the whole back-pressure on the piston when the engine has a rate of thirty miles an hour gives a loss of about $\frac{1}{50}$ of the load.

The area of the blast-pipe to produce a certain draught depends on the grate area, area of the cross-sections of the tubes, area of the chimney and the capacity of the smoke-box; of these the ferule area at the fire-box most affects the blast-pipe. With a small chimney a small orifice is unnecessary, the area of the best chimney seeming to be $\frac{1}{15}$ that of the grate, and its length being more than four times its diameter.

The capacity of the smoke-box, in cubic feet, is usually three times the number of square feet in the area of the grate.

In even the worst-constructed boilers the area of the blast-pipe is greater than $\frac{1}{90}$ of the area of the grate, and in ordinary boilers it is $\frac{1}{55}$.

Hence, in all cases it is possible to get a wide enough blast-pipe for the escape from the cylinder by having sufficiently large grate area and sectional area of tubes.

209. **Safety-Valves, &c.**—Of these there are two on every boiler, both usually placed on the steam dome, though often separate, as in Fig. 56. It is better that they should be kept away from the mouth of the steam-pipe, to prevent priming in the cylinders when the valves blow off. Safety-valves of locomotives are constantly in action, and they never gag in their seats.

A spring attaches the end of the lever to a stud in the boiler-plates, and this spring acts as the weight acts in the safety-valve (Fig. 51). It is important that the opening for escaping steam should be great on a slight lifting of the valve, and therefore that the elasticity and scope of the spring should be great.

Direct action safety-valves are more or less like that

shown at S (Fig. 56), and they are important improvements on valves whose springs act at the ends of levers; for they give immediate relief to excessive pressure, by the ample opening for steam given by the slightest lift of the valve.

Bourdon's is the pressure-gauge usually attached to locomotive boilers. A glass water-gauge and three gauge-cocks are commonly applied. Whistles like T N (Fig. 56) are of brass, the bell being 3 inches in diameter, and 2½ inches deep.

210. **Priming.**—Enough has been said about the causes and the methods of prevention of priming. We find by practice that the average depth of the steam-space in the boiler, or *steam-chest* as it is called, ought to be one-fourth of the diameter of the barrel. Under these circumstances, an evaporation of five cubic feet of water per cubic foot of steam-space per hour may be effected without much priming.

The steam-space in the fire-box shell is equal to the steam-space in the barrel. The boiler must be frequently cleaned or blown off.

Independent means of separating water and steam in the pipes are sometimes useful.

211. **Consumption of Fuel in Locomotives.**—Of this subject Art. 142, Book II., treats very fully, and the student there worked exercises to find the weight of water at 100° converted into steam by 1 lb. of fuel, when the number of square feet of heating surface per square foot of fire-grate, and the number of pounds of fuel used per hour per square foot of fire-grate, were known.

CHAPTER VII.

STABILITY OF THE LOCOMOTIVE.

212. THE stability depends on the internal arrangement, the balancing of moving masses and the distribution of the load. When the internal disturbing forces are balanced and the extreme wheels properly placed, great liberties may be taken with the general arrangement. Inside and outside cylinder engines may be made equally steady, the centre of gravity may be high or low, and the driving-wheels may be in any position we please.

213. **Internal Strains.**—The great internal strains in locomotives are produced by the centrifugal force of cranks, connecting-rods and eccentric discs, allowed to be communicated to the framing, producing rapidly changing vertical and longitudinal stresses; by side-pressure on the slides due to the use of short connecting-rods producing vertical stresses which change in nature every stroke; by inequalities in the admission of steam to the two ends of the cylinder; by the above stresses being communicated from two engines to the same framing; and by the ordinary stresses due to steam-pressure in the cylinder.

These last strains are sufficiently guarded against by making the framing as rigid as possible. The vertical motions are insignificant compared with the longitudinal when the wheel base is considerable, when there are many points of support to the frame, and when the springs, particularly the fore and end springs, are as strong and stiff as possible. If these springs are easily strained, accidental vertical motions given by irregularities in the rails, &c., may coincide with the internal

vertical motions and increase the oscillations. These oscillations show themselves in *pitching*.

A *longitudinal fore and aft* oscillation is given by the momentum of the reciprocating masses being alternately added to that of the frame and taken away again. It is often very considerable. It is much increased by inequalities in the admission of steam to the two ends of the cylinder, and is very evident when the total momentum of the engine is small, that is, when approaching a station. Employing strong springs between the engine and tender gives a partial remedy. A pencil placed on one of these springs will give indications of the oscillation.

We will now consider the SINUOUS MOTION produced by centrifugal force and changes in momentum of the moving masses.

214. **Balancing of Engines.**—When a body moves in a curved path it has a tendency to leave that path, and this tendency is called centrifugal force. The centrifugal force of a body moving in a circular path is measured by the expression $\frac{m\,v^2}{r}$, where m is the *mass*, v the velocity of the body, and r the distance from its centre of gravity to the centre of the circular path. This is better stated in the form—

$$m\,r\,\omega^2$$

where ω is the angular velocity of the body (circular measure).

It is sometimes necessary to find a mass to revolve round a shaft in such a way that its centrifugal force is equal and opposite to some other force, such as the momentum of moving masses at the crank-pin of a steam-engine, or perhaps equal and opposite to the centrifugal force of some body revolving about the same axis, such as the crank or the eccentric disc.

When two masses are directly opposite to each other on a shaft, their centrifugal forces may be made to

balance. When not opposite they cannot be made to balance, but two masses may balance one opposite mass placed between them. We are evidently here concerned with parallel forces acting at right angles to the shaft.

It may be shown that when the centre of gravity of a number of revolving bodies is in the axis of rotation, the pressure produced by centrifugal force on one side of the axis is equal to the pressure on the opposite side. Sometimes these equal pressures are not opposite to each other, although parallel, and hence they produce a couple which tends to alter the position of the axis or shaft, producing pressures at the bearings. The pressure at a bearing is evidently the moment of the above couple divided by the distance between the bearings.

When there is no tendency to change the direction of the axis, it is said to be a *permanent axis*. All axes of rotation in a steam-engine ought to be permanent axes. When this is the case, and when the engine is suspended by two points on the axis and made to work, there are no visible oscillations. M. Chatelier experimented on the stability of balanced locomotives by suspending them in this way.

There are two separate sets of forces acting on the crank-shaft which have to be balanced.

(1.) The centrifugal force of the crank, crank-pin, and as much of the connecting-rod as may be supposed to follow the path of the crank-pin (say one-half of it). The mass, or weight, of each of these multiplied by the distance of its centre of gravity from the centre of motion divided by the length of the crank gives the mass which, on the crank-pin, would produce the same centrifugal force. Let this be called m.

In designing engines it is well to make an exact calculation of the nature of the pressure on the crank-axle due to the connecting-rod by considering the curve traced out by its centre of gravity. When this is not done we consider half the rod to be collected at

the crank-pin, the other half to be moving along with the piston.

Let R = length of crank, ω its angular velocity.

At the end of the stroke, when the horizontal component of the centrifugal force is greatest and the vertical component vanishes, the horizontal pressure on the axle caused by centrifugal force is—

$$m \, R \, \omega^2$$

(2.) The pressure produced by change in the momentum of the second half of the connecting-rod, the piston, piston-rod, and slide.*

Let the total *reciprocating* mass be M.

It may be shown that the *loss of momentum is most rapid* just at the end of the stroke, and is then $M \, R \, \omega^2$ per unit of time, hence the horizontal pressure on the axle is—

$$M \, R \, \omega^2$$

Now, a mass m^1, or masses, whose sum is m^1, may be placed on the driving-wheel, or wheels, at a distance r from the axis such that the centrifugal force of m^1 may be equal to the sum of the above pressures. Thus—

$$m^1 \, r \, \omega^2 = M \, R \, \omega^2 + m \, R \, \omega^2$$

or

$$m^1 \, r = (M + m) \, R$$

from which, when r is assumed, m^1 may be calculated.

For the axis to be permanent in inside cylinder engines, m^1 must be divided into two parts, one for each wheel, inversely proportional to the distances from the wheels to the crank.

For outside cylinder engines we get masses for the two wheels, whose *difference* is m^1, the masses being as before

* In coupled engines the coupling-rods and cranks have to be separately considered.

proportional to the distances from the wheels to the crank in question.

Thus, a consideration of each cylinder gives two balance weights, one usually very much smaller than the other. As the cranks are at right angles to one another, the masses will be a quadrant apart on each wheel. Instead of these two masses on the wheel, we may employ one mass somewhere between the two, such that its centrifugal force is the resultant of theirs. If the two masses are a and b, then it is evident that if $c^2 = a^2 + b^2$, then c is a mass which may be substituted for a and b. If $\tan \theta = \frac{a}{b}$, then θ is the tangent of the angular distance between c and b.

It often happens in outside cylinder engines that the distance from the wheel, or centre of gravity of balance weight, to the crank or centre line of the cylinder is very little; the corresponding balance weight for the other wheel is therefore very small, and may even be neglected. In inside cylinder engines it will be found that, whereas the cranks are at right angles to one another, the balance weights on the two wheels on the opposite side of the axis to the cranks are often only 50° apart.

In inside cylinder engines with coupled wheels, the outside coupling-rods and cranks are usually made to balance the inside moving parts. These engines work very smoothly indeed.

Outside cylinder engines with coupled wheels are very unstable, from the use of small wheels requiring very rapid revolution of the crank-axle; from the cylinders being farther apart than usual that the coupling-rods may have room; and from the number of reciprocating parts being increased. The conditions seem to admit of no remedy for these defects.

It has been shown by experiment that the application of suitable balance-weights is attended by a sensible reduction of resistance on the rails at high speeds. Engines unbalanced cannot attain as high speeds as when balanced, with the same consumption of fuel.

The balance-weight ought to be distributed over two or three of the spaces of the wheel that the stress on the tire may be small.

The reciprocating parts of engines ought to be as light as possible, and the width of cylinders as nearly as possible equal to that of the wheels.

Practical rules for balance-weights :—

R = length of crank.

r = distance of centres of gravity of balance-weights from centres of wheels.

e = distance apart of centre lines of cylinders.

d = distance apart of the wheels or centres of gravity of the balance-weights.

θ = angle which position of centre of gravity of balance-weight makes with *near* crank.

w = total weight of crank (referred to the pin), pin, connecting-rod, piston, and rod.

(1.) Inside cylinder engines with uncoupled wheels—

$$\text{The balance-weight} = \frac{w\,R}{2dr}\sqrt{2\,d^2 + 2\,e^2}$$

$$\tan\theta = \frac{d-e}{d+e}$$

(2.) Outside cylinder single engines with uncoupled wheels—

$$\text{The balance-weight} = \frac{w\,R}{r}.$$

$$\theta = 180°.$$

The balance-weight to be exactly opposite to the crank.

(3.) Inside cylinder engines with wheels coupled—

Find by rule (1) if the weight of the coupling-rods, &c., is too great. If so, let counter-weights equal to the difference be placed opposite the outside cranks. If too small, the difference must be made up with balance-weights as in Rule (1). The position of the outside cranks is found by Rule (1).

(4.) Outside cylinder coupled engines—

Find revolving weight of coupling-rods, &c., for each wheel. Also find sum of the weight of the piston, rod, slide, and half connecting-rod. Divide this latter among the wheels, adding the given revolving weight already on them. Let this be used on each wheel according to Rule (2).

CHAPTER VIII.

FRAMING, ETC.

215. **Engines with Inside Cylinders.**—BURY saw (Art. 187) that with outside frames, when the cylinders were placed in the smoke-box, the great stresses from the cylinder to the crank-shaft were not taken up by direct connection, but were made laterally by the smoke-box stays. He designed a more direct and better connection in his *inside framing*.

SHARP chose merely to strengthen the smoke-box stays by an accumulation of bars and riveting, and he still retains the outside framing.

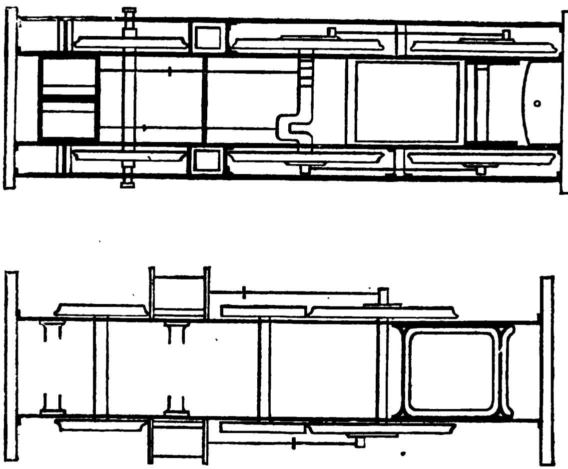
The ends of the crank-shaft are carried by the outside framing, and although inside bearings are also provided, they are weak, and practically of little value.

With the outside frame we consider the bearings as being outside the wheels, and with the inside frame we consider the bearings as inside the wheels, and between the wheels and the cranks. Obviously this latter is the better position for a bearing, as the strain transmitted from the crank through the bearing to the wheel is thus transmitted through a shorter length of the shaft.

Now in an engine with outside frame, the bearings inside the wheels must wear most, as they are nearer the cranks; and the only way to remedy the evils arising from this, is to preserve the outside frame as before for the sake of general stability, but to let the principal bearings for the crank-shaft be placed on an inside frame between the wheel and the crank, this frame being continued in a rigid manner from the back

of the fire-box as far forward as the buffer-beam, right through the smoke-box. In coupled-wheel engines, all the powerful bearings are placed in the inside frame, for obvious reasons.

Fig. 64 shows an arrangement of framing practically perfect.



FIGS. 64 and 65.

The two frames are very strong and give plenty of room for springs, &c., and great bearing for cylinders (Art. 191). An inside frame alone is sufficient for engines under twelve tons.

The application of these principles of stability to *outside cylinder* engines is obvious, and the sketch of the framing (Fig. 65) needs no explanation.

For quick speeds there must be supports at the ends of the engine and yet a short-wheel base must be maintained.

One-fourth of the whole weight is sufficient for the leading wheels, and they are best placed when their axle is within a foot of the smoke-box tube-plate. The driving-axle is best in front of the fire-box, where it will get most of the weight, and hence will be in its best position for traction. Thus, an extra pair of wheels is necessary to be placed behind the fire-box, and to support $\frac{1}{12}$th of the weight of the engine. This is the six-wheeled engine.

It may be shown that 18 tons is the greatest weight proper for a six-wheeled engine with uncoupled wheels, except when speed, not drawing-power, is of the greatest importance.

When wheels are coupled, the weights supported by them should be equal. Six-wheeled engines, with the two hind axles coupled, seem best for high speeds, and are certainly most suitable for outside-cylinder engines.

For curves, the driving and leading wheels of uncoupled engines should have flanges $1\frac{1}{4}$ inch thick, and the hind wheels should be plain when the hind load is under two tons, or have thin flanges with much clearance from the rails when the hind load is over two tons.

When cylinders are outside and are overhung in front, the leading wheels may be $3\frac{1}{2}$ feet in diameter, with 6-feet drivers, increasing in diameter as the drivers increase. Their axles must be very firm. The hind wheels when uncoupled may be practically of the same diameter as the leaders, but their axles must possess some flexibility.

216. **Crank-Shaft.**—As we have said, it is usual with inside cylinders to have two bearings for the crank-shaft, placed between the wheels and cranks, and two less important bearings on the outside framing. These less important bearings often may be left out altogether with great propriety.

The cranks are forged in one piece with the shaft, in positions at right angles to one another. The crank-pin is made very strong, and of as great a diameter as possible

In Fig. 38, showing half of a crank-shaft, A B is the centre line of the engine, and C and D are the edges of two eccentrics.

217. **Axle-boxes and Springs.**—An axle-box is one of the bearings for an axle, and consists of a cast-iron block containing two halves of a bearing, clasping the turned part of the axle. This block is compelled to move up and down in *guards* on the *frame* of the engine, and it is pressed upwards against the rod A

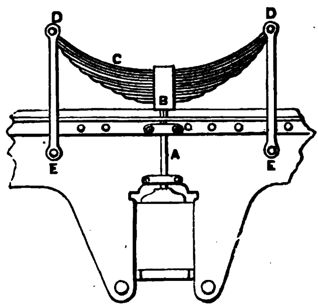


FIG. 66.

(Fig. 66). When some inequality in the rail pushes the wheel upwards, the axle-box is pressed upwards, and this pushes the rod A. This rod conveys the pressure to the box B, and tends to flatten the compound steel spring C, which communicates the upward push by the rods D E to the framework at E. Thus a push upwards is never transmitted suddenly to the framework from the wheels, but in a very gradual manner, so that the boiler suffers nothing from a slight inequality in the rails. A spring should yield equally in every part, and therefore a compound spring

like that shown in Fig. 66 is made thick in the middle.

The span of the spring is the distance D D, and this increases slightly with increase of load.

Let n = number of plates.
t = thickness of each plate.
b = breadth of plates.
s = span.
L = load at centre.

Then strength of spring $\propto \dfrac{t^2 b}{s}$.

Deflection $\propto \dfrac{L s^3}{n b t^3}$.

In practice the deflection is about half an inch for every ton of load. Leading springs should be least flexible. The plates are kept fair by studs and slots at their extremities.

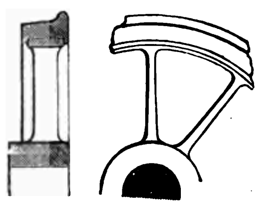


FIG. 67.

218. **Wheels.**—A wheel consists of the *nave* in the middle, the *spokes* or arms, the *rim*, and the *tyre* outside all. Wheels are usually of wrought iron throughout, the spokes being welded together to form the nave; but indeed a cast-iron nave is often used for low speeds. The rim is usually about one inch thick and five inches broad. The wrought-iron tyre is about two inches thick (Fig. 67), and is *bored* out

how sudden, and ought, therefore, to have made a careful study of his engine.

When the parts of one engine get deranged, it may perhaps be disconnected from the other. One engine may perhaps draw the train or a part of it ahead, or the engine by itself may go ahead for assistance.

The driver is understood to remain on the foot-board at all times, within reach of the handles. He carefully attends to the feeding, opens the pet-cock whenever necessary, shutting off the feed-valve in going up inclines and in leaving stations, and opening at intervals on approaching a station. He charges the fires with small quantities of coal at a time on approaching a station, during the short intervals when there is no feed-water entering. The damper is partially closed and the blast-pipe opened, the regulator being completely closed when approaching a station and within half a mile.

The train is stopped by means of the brake. In some cases the engine may be reversed, but as this produces great stresses in the various parts, and as cinders may get into the cylinder, it ought to be avoided if possible.

Blowing-out may be sometimes necessary to prevent priming if the water is dirty. The boiler ought to be carefully cleaned out three times a week.

The fires are allowed to get low towards the end of a journey. Before the engine cools down, let the parts be cleaned down and examined carefully, the tubes cleaned, and the smoke-box cleared of ashes.

The brasses, axles, stuffing-boxes, and valve-gear generally are most apt to have become deranged. (See *Bourne's Catechism.*)

CHAPTER X.

PERMANENT WAY, ETC.

221. As this subject really belongs to another province of engineering, we shall be brief. The great expense of railway construction arises from the per-

manent way; the cuttings and embankments, the draining, the ballasting, and then the laying of sleepers to bear chairs and rails.

In England, great care is usually taken with the permanent way, and so English railways cost a great deal at first, but are not very expensive to keep in order; whereas in America, where new countries were opened up by most of the lines of railway, first cost was made a minimum, and, as a consequence, the cost of maintenance is very great. The great problem of the day in railway engineering is to make the first cost moderate, as well as the current expenses.

The Formation Level is the surface of the earthwork on which the ballast and sleepers lie. Under this head it is usual to consider the side slopes, drainage, retaining walls in cuttings, &c.

It may be shown that American railways are deficient in drainage, and in the solidity of the formation level, and that from these deficiencies arises much of the increased current expense.

Ballast consists of broken stone or brick, sand, gravel, or cinder. Its office is to distribute the load over the whole surface of earthwork, and it seems to fulfil this office best when about two feet deep, tightly enclosing the sleepers, and allowing drainage through its mass.

The road ought to be elastic and yielding, not rigid, as if of hard rock; and yet possessing a certain amount of stiffness.

There are inequalities in all kinds of rails, and shocks are continually being given. In a rigid road there is no yield when a shock comes, and from a knowledge of strength of materials we see that this condition is very destructive. Again, a road which gives too great a yield is also bad, and hence the very greatest attention should be paid to ballasting.

222. *Gauge.*—The various gauges (or inside distances between a pair of rails) employed in the kingdom are the following:—

1. The most usual gauge, 4 feet 8½ inches.
2. The Scotch gauge, 5 feet 6 inches.
3. The Irish gauge, 6 feet 2 inches.
4. The Great Western gauge, 7 feet.

223. *Rails* are supported either on transverse sleepers, with a support at every sleeper; or on continuous longitudinal bearings. Sometimes the rail is supported by blocks of stone instead of the sleepers, but such a system gives insufficient elastic yielding, and presents great difficulties when the road requires mending. Sometimes isolated blocks of wood are used instead of these blocks of stone, and answer much better.

Rails are of different shapes, as they are intended for one sort of support or another. The rail with the **I** section (double-headed), and that with the ⊥ section (flat-bottomed) seem to be most generally approved of.

The double-headed rail is more employed when first cost is unimportant. It requires many chairs. The unsupported joint in which a fish-plate embraces the lower flange seems best for this rail.

Flat-bottomed rails cannot be made with wide enough bottom flanges for the stability expected from them.

The *chair* for the above rails is a casting which may be bolted to a sleeper. It has a space in its upper side, in which the rail may be tightly held by means of a key of wood or iron.

On the Continent, cast iron is often used for supporting the rails, in transverse beams, or in detached castings, like dish-covers. Perhaps the flat-bottomed rail is better in this case.

The Sandwich sleeper is a large block of wood, supporting the rail all along its side at the upper corner, and connected with the block supporting the opposite

rail by straps of wood or iron, to preserve the gauge. The bearings are long, the necessary elasticity is provided, and the blocks are cheap.

It is seen that the sleeper should give up its load to the ballast at points directly under the rails; but transverse sleepers cannot do this, and they loosen, and with their great length gradually begin to vibrate and beat the ballast to powder as a train passes over them.

The longitudinal system has long been established on the Great Western Railway, and it is seen there that the rails necessary for longitudinal sleepers spring under the passage of a train, because of their want of lateral stiffness.

A light rail is always made of better iron than a heavy one, as it is much better worked in the rolling-mill; and hence light rails of about 50 lb. per yard are coming into use everywhere, and are thought to possess great durability.

For England, where ordinary sleepers are so much used, the **I** rail seems best fitted, as with a chair on every sleeper the rail possesses great lateral stiffness, and as it is deep it deflects very little in the distance from sleeper to sleeper. (Deflection of a beam varies inversely as the cube of the depth.) That the rail may be as strong as possible, the head of the rail should be larger than the bottom in the ratio 15 : 11, since this is the ratio of the powers to resist extension and compression of wrought iron.

One hundred miles of wooden railroads of 4′ 8″ gauge are in use in Canada. Rails of maple 7″ × 4″ are notched into cross sleepers 8″ square and 20″ apart. There is less adhesion in wet weather than is really necessary for ordinary inclines. This system may yet be employed in flat districts where the traffic is less than usual.

224. **Joints.**—Rails are usually either 18 or 21 feet long, as it is not thought safe to make them longer

in an ordinary rolling-mill, and so *rail-joints* are numerous on a railway.

All railroads are to some extent rough, and give the wheels an irregular motion. If the road is not firm under the rails, a rigid joint is productive of good; but if the road is very rigid, the joint must allow of a little movement on the occurrence of a sudden shock. As far as all parts of the rail—except the joint—are concerned, the sleeper gives quite enough elasticity; but at the joint we must have more yielding than at a bearing on one of the sleepers. To do this and preserve the same level in the united rails, so as to

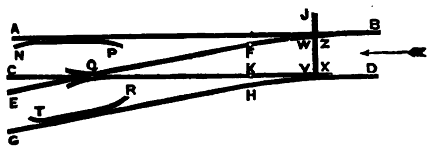


FIG. 69.

reduce the shocks as much as possible, and at the same time make a strong connection, is the office to be fulfilled by a good *rail-joint*.

When we come to consider this question, we see that the levels can be maintained only by a thoroughly spliced joint; and as low rails cannot be spliced properly without introducing a destructive rigidity, they must be left out of consideration for the future.

When the rail is deep, and the under side of the head is moderately square, the fish-joint is cheapest and best. It is usually introduced between two sleepers placed two feet apart, and consists of two plates of wrought iron 18 inches long, and less than an inch thick, one on each side of the rail, bolted together through the rail by four bolts.

225. When it is necessary at any point of a line of rails to move a train to one of two or more lines of rails meeting at the point, and having the original line as their common continuation, a *switch* is made use of. Fig. 69 will give an idea of a simple switch. The two hinged parts F W, K V are rigidly attached to the bar J W V, which, by sliding backwards or forwards, causes a *point* to bear against B Z or D X. In the position shown in the figure, a train leaving B D would move along B E G D.

The points W V usually slide on an iron supporting table.

To the left of the figure an ordinary crossing is shown.

We do not here mean to give more than the merest introduction to this subject.

226. **Centrifugal Force at Railway Curves.**—All moving bodies tend to move in a straight line, and hence a body moving in a curve is continually being constrained by forces at right angles to its path. The centrifugal force, acting horizontally at right angles to the direction of the rails, is measured by the expression—

$$\frac{\text{Weight} \times \text{square of velocity in feet per second}}{32{\cdot}2 \times \text{Radius of curve.}}$$

Now this force is very considerable in sharp curves, or curves of small radius, and the outer rail has to be raised to prevent the train from falling over sideways; but with a curve of great radius, the conical inclination of the wheel tyres increases the radius of the outer rolling surface, and sufficiently counteracts the centrifugal force, without our raising the outer rail.

When the rail has to be elevated, the amount of elevation is given by the formula—

$$\frac{a\,v^2}{32{\cdot}2}\left(\frac{1}{R} - \frac{4\,n\,m}{d\,a}\right)$$

Where d is the outer diameter of wheels.

„ a is the gauge of the rails.

„ n is the inclination of the tyre in circular measure; that is, the inclination in degrees divided by 57·3.

„ v is the velocity of the train in *feet* per *second*.

„ m is the deviation or lateral play of the wheels, as a fraction of a foot.

227. **Resistance to Traction.**—Certain experiments conducted on a straight line of broad-gauge rails newly laid down led to the following rules. Unfavourable circumstances must increase the resistances here found:—

1. To find the total resistance of a train, engine and tender, square the speed in miles per hour, divide by 171, and add 8 to the quotient; the result is the resistance in pounds per ton of the moving weight.

2. To find the resistance of the train alone, square the speed in miles per hour, divide by 240, and add 6 to the quotient for the resistance in pounds per ton of the moving weight.

A curved line adds 12 per cent. to the resistance on broad-gauge rails, and 20 per cent. to the resistance on narrow-gauge rails.

The wetness of rails adds very little to the resistance, perhaps not more than 2 lb. per ton of load, while an imperfect road may often add as much as 40 per cent.

Strong side-winds add as much as 10 per cent. to the resistance on broad-gauge rails, and 30 per cent. on narrow-gauge rails.

228. **Narrow-Gauge Railways.**—These are intended as second-class railways, where lower speeds may be employed, and where the accommodation need not be so good as on the main lines. On the other hand, the first cost and working expenses will be much smaller than on the broad gauge. The waggons for transporting goods will be of a smaller size, and hence will be more

suited to changes in the rate of traffic. Again, they will in other ways be much better suited to the requirements of the traffic to be met with in thinly populated districts. The engines may be made powerful without damage to the light rails, as they move slowly, and may be provided with a great number of wheels.

From the published cost of constructing permanent way with narrow gauge, and from a consideration of published statistics showing the accommodation afforded by the use of this system, it may be shown that as a general rule the most efficient gauge is that which was adopted for the Indian lines, namely, one metre, or 3 ft. $3\frac{3}{8}$ in. As the gauge is made smaller, the loss in accommodation is more rapid than the lessening of the expenses.

Mr. Fairlie and Mr. Spooner recommend smaller gauges than this, however.

To spread a powerful engine on a great number of wheels, and at the same time to maintain the power of passing round curves, was long felt to be a difficulty. Coupled engines are difficult to manage, giving great uneasiness to one engine-driver.

In narrow-gauge lines it becomes necessary to spread a heavy engine over a great number of wheels.

The *Meyer* locomotive has one large boiler carried on two bogies, the coupled wheels of each bogie being driven by two cylinders. In one representative case for the narrow gauge the boiler is 18 feet long, 1′ 11″ diameter; the diameter of each cylinder is $17\frac{5}{16}$ inches, the wheel base of each bogie is 8′ $8\frac{3}{4}$″, and the total wheel base 27′ 8″. A Meyer locomotive exhibited at Vienna is described in the engineering papers.

Fairlie Engine.—Two bogies with coupled wheels, each bogie provided with two cylinders, are connected by means of a peculiar framing carrying two boilers (each with an independent chimney), whose smoke-boxes are adjacent. The steam-pipe is coiled, like a

spiral spring with one thread, in the smoke-box, and yields when the engine is passing round a curve. This engine is capable of passing round curves of 170 feet radius with a considerable speed. It is powerful, and well-fitted to work on steep inclines. It runs in either direction. These qualities of the Fairlie engine are most strikingly shown on the Festiniog Railway, on which, too, the safety of a very narrow gauge with steep inclines and many curves, was first demonstrated. The Fairlie engine employed on this railway (2 feet gauge) has a total wheel-base of 19′ 1″, the wheel-base of each bogie being 5 feet, wheels 2′ 4″ diameter, cylinders 8¼″ diameter, stroke 13″, grate area 11 square feet, heating surface 730 square feet.

The first Fairlie engine was, I believe, *The Progress* (Dec. 1865), with 15 inch cylinders, and whose weight when charged was 42 tons. Since 1865 these engines have prevailed against much prejudice.

The locomotive *Seraing*, made for the Simmering Incline in 1854, had somewhat the same general construction as the Fairlie engines, now coming into general use. The use of the *Seraing* was for some reason discontinued. Again, on the Northern Railway in France, four-cylinder passenger engines, their wheels separated into coupled groups, had been working before Mr. Fairlie constructed his engine.

BOOK IV.

MARINE ENGINES.

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CHAPTER I.

RESISTANCE TO THE PROGRESS OF VESSELS.

229. WHEN the vessel has attained a uniform speed the resistance which must be overcome by the propeller in maintaining this speed consists of that due to: (1) The distortion of particles of water; (2) the production of currents, eddies, and foam; (3) the production of waves; (4) the production of a current by friction.

(1) May be neglected in the case of motion of a large body through water. It has a considerable value when small models are employed for illustration.

(2) May be neglected for moderate speeds in a well-formed vessel, or one built on Scott Russell's *wave-line* system.

(3) A well-formed vessel creates two, and only two, non-diverging waves when the speed is moderate. Scott Russell found that the resistance due to this cause was insensible at "moderate speeds," but that this is not the case when the speed exceeds that at which a wave naturally travels whose length bears a certain fixed proportion to the length of the after-body of the ship.

For any vessel there is a certain limit of speed below

which the resistance due to the production of waves is inconsiderable, and above which the resistance increases rapidly with the speed.

When the vessel is well formed, *Scott Russell's rule to find the greatest speed* in knots, suited to a given length of after-body in feet, is—take the square root of $2\frac{2}{3}$ times this length.

We shall always suppose that ships move with "moderate speeds." The best definition of moderate speed is—that speed at which the H.P. necessary to overcome the resistance varies as the cube of the speed. (See below.)

(4) It may be shown that at *moderate speeds* the resistance due to the production of frictional eddies by a given portion of the skin of the vessel varies as the square of the speed and the cube of the ratio which the velocity of the neighbouring water with respect to the part in question bears to the speed of the vessel.

The resistance of a vessel then is almost altogether due to friction. By friction here we mean friction among the films of water which have to be set in motion, and which get their motion from each other. Unless when the velocity is very excessive, one film will adhere to the surface of the vessel, retaining the velocity of the vessel, films outside this moving at less and less velocities, as their distances from the ship are greater. Hence the friction is greatest at the bows, and after a time further additions to the length of a ship have no great effect in increasing friction, since the surrounding water no longer requires to be set in motion. It is evident from this that the vessel surrounds itself with a current of its own making. If this current is very great, since it is in a direction opposite to the direction of motion of the paddle-wheel floats, and since it is in a direction opposite to that which the water would receive from a screw-propeller, the *slip* of the paddle, or screw, may be materially diminished, reduced to zero, or even made to have a negative value. It will be found that there is a great gain in speed in following in the wake of another vessel.

The forward velocity of the water due to friction has to be taken into account in considerations of the efficiency of screw-propellers. It may be calculated approximately (*see* Rankine's *Shipbuilding*, p. 249), or determined experimentally.

230. **Computation of the resistance in pounds.**—Rankine's method proceeds by deducing the frictional resistance from an approximate value of what he calls the *augmented surface* of the vessel. It is therefore applicable only to those vessels in which the resistance due to frictional eddies forms the whole appreciable resistance; but such is the case with all vessels of proportions and figures well adapted to their speed, and as for misshapen and ill-proportioned vessels, there does not exist any theory capable of giving their resistance by previous computation.*

At moderate speeds, that is, when the speed never exceeds that given above by Scott Russell's rule, to find the *augmented surface*—

$$A = m\ l\ k$$

where A is Rankine's *augmented surface.*

m is the mean immersed girth measured on the body plan of the vessel. The immersed girth at any section is the length of a string which would stretch from a point on the water-line on one side of the vessel to an opposite point on the water-line on the other side.

The *mean* immersed girth is found as the mean pressure is found from an indicator diagram.

l is the length of the ship on the water-line.

k is a co-efficient = 1 + four times the mean of the squares of the sines of the maximum angles of obliquity of stream lines.†

The augmented surface here found in square feet multiplied by the square of the speed in knots per hour and by a constant co-efficient K gives the resistance in pounds.

K for clean painted iron vessels = ·01.
,, ,, coppered vessels = ·009 to ·008.
,, moderately rough iron vessels = ·011 and upwards.

* Rankine's *Shipbuilding.*

† There is a certain tangent to a stream-line which makes a *maximum* angle with the axis of the vessel; four times the mean of the squares of the sines of all such angles is here taken.

231. **Approximate Computations of the Resistance.** —In vessels of exactly the same shape, *the augmented surface* is proportional to *the area of the greatest immersed cross-section* of a vessel, and the square roots of these are proportional to the cube root of the *displacement* in tons. This is *approximately* true for all well-shaped vessels, and hence the following rules :—

(1) The resistance in pounds is equal to the square of the speed in knots per hour multiplied by the square of the cube root of the *displacement* in tons, and also by a constant varying from ·8 to 1·5 for different steamers.

(2) The resistance is calculated from the *greatest immersed section* in exactly the same way as from the *augmented surface*, K being different. In future the resistance will generally be represented by the expression K A v^2, where A is either the augmented surface or the immersed section; K being the proper constant. Hence K A is a known constant for a particular vessel.

The results obtained by using the augmented surface are much more correct; but engineers are better acquainted with the approximate methods, and we shall employ the immersed section in Chap. V.

232. **The H.P. effectively employed in Propulsion** is obtained by multiplying the resistance by the speed in knots, and dividing by 326. (This is easily seen from the principles of Book I.)

233. **The Indicated H.P.** corresponding to a given speed is obtained by dividing the effective H.P. just found by the efficiency of the machinery conveying the motion from the piston to the vessel—that is, of the engine machinery and of the propeller. This efficiency generally varies from ·6 to ·625 for good propellers. Rankine takes ·613 as an average. So that to get *the indicated* H.P. we take resistance in pounds × speed in knots per hour ÷ 200.

Example. – Taking six water-lines and the keel, Rankine finds k in H.M.S. *Warrior* to be 1·275. He finds the mean immersed girth to be 76·3 feet. The length on the plane of flotation is 380 feet; hence the *augmented surface* is 36979 square feet. The resistance is $36979 v^2 \times \cdot 01$, and the H.P. required for propulsion $= \dfrac{36979\, v^3 \times \cdot 01}{33000}$.

If the efficiency of the machinery is taken at ·613, then the indicated H.P. $= 1 \cdot 849\, v^3$.

The indicated H.P. on trial was 5471, therefore the speed, as calculated from the above formula, was 14·35 knots. Now it was found that during the trial the speed was 14·354 knots, so that Rankine's rule may be depended on. Other examples will be found in *Shipbuilding*, the mechanical journals," Oct. 1861, and the "Philosophical Magazine" (Great Eastern), April, 1859.

Rankine's result has been successfully employed in determining the probable speed of the *Great Eastern*, and of many other steam vessels.

234. The resistance of a vessel increases in rough water (1) from revolutions of particles of water, (2) from alterations in the immersed figure of the vessel.

235. We see (Art. 232) that the H.P. required for *propulsion* at moderate speeds is proportional to A v^3, or to $D^{\frac{2}{3}} v^3$, where A represents either the augmented surface or the immersed mid-ship section, D the displacement of the vessel in tons, and v the speed. Again, if we assume that the efficiency of the propeller and machinery is the same at all speeds (*see* Arts. 233, 243, 261), the *indicated* H.P. is proportional to A v^3, or to $D^{\frac{2}{3}} v^3$, and we get the following definitions :—

(1.) The expression $\dfrac{D^{\frac{2}{3}}\, v^3}{\text{H.P.}}$ has a definite value for every steam vessel; and in different well-shaped vessels it represents the relative amounts of work usefully performed by 1 lb. of fuel, or by one indicated H.P. of the steam engine. It has been called the

Locomotive Performance. It ranges from 200 to 260 in good vessels, and is sometimes as low as 150 in bad examples.

(2.) The expression $\frac{A\,v^3}{\text{H.P.}}$ has been called by Rankine the *co-efficient of propulsion*, A being the augmented surface. For well-shaped clean iron vessels, this co-efficient may be taken at 20,000. In obtaining this, it is assumed that the efficiency of the machinery (Art. 263) is ·613.

Rankine's co-efficient is only slowly coming into use. We give some exercises on the *Locomotive Performance.*

Exercises.

(1.) A ship, whose displacement is 1000 tons, has a speed of 12 knots when the indicated H.P. of the engines is 850: What is the locomotive performance?

$$Ans.\ \frac{1000^{\frac{2}{3}} \times 12^3}{850}, \text{ or } 203{\cdot}3.$$

(2.) How fast will the vessel of Ex. (1) proceed when the indicated H.P. is 425?

$$Ans.\ \frac{1000^{\frac{2}{3}} \times v^3}{425} = \frac{1000^{\frac{2}{3}} \times 12^3}{850}$$

Or

$$v^3 = \frac{12^3 \times 425}{850}$$

which gives $v = 9{\cdot}5$ knots.

(3.) When the indicated H.P. is 1200, the speed is found to be 9 knots: What is the H.P. when the speed is 12 knots?

$$\frac{D^{\frac{2}{3}} \times 9^3}{1200} = \frac{D^{\frac{2}{3}} \times 12^3}{x}$$

Or

$$\frac{9^3}{1200} = \frac{12^3}{x}$$

which gives $x = 2844{\cdot}4$ H.P. the answer.

236. **Speed against a Current.**—To find the speed of a vessel opposed by a current, which will give the best economy of fuel. We still assume that the efficiency of the propeller is the same at all speeds (*see* Arts. 233, 243, 261).

Let v = velocity of current in miles per hour. Let x = velocity of the ship due to the coals used; hence, $x - v$ = distance in miles passed over every hour by the ship.

Now, the consumption of fuel per hour $\propto x^3$ and the consumption per hour for every mile passed over $\propto \frac{x^3}{x-v}$; and this must be as small as possible for the best economy of fuel. By the Differential Calculus this is a minimum when $2x = 3v$; hence the ship ought to use as much coal as would, if unopposed by a current, give it a velocity half as great again as that of the current; or, as it is commonly put, *the ship ought to steam half as fast again as the opposing current.*

237. **Fuel for a Voyage.**—Let c be the consumption of fuel in one hour. If a is the distance between two places, $\frac{a}{v}$ is the time of a voyage between these places.

Now, $c \propto v^3$, hence $c\frac{a}{v} \propto a v^2$. Now, $c\frac{a}{v}$ is the whole consumption of fuel in the voyage.

Hence the consumption of fuel in a voyage between two places, varies as the distance multiplied by the square of the velocity.

238. **Experiments.**—In experimenting on the speed of a vessel, when there are tidal currents, it is neces-

sary to make three runs, two *directly* with or against the tide and one opposite to these two. The results of these three runs enable us to eliminate the effect of the tide in an approximate manner. (*See* Rankine's *Shipbuilding.*)

CHAPTER II.

PADDLE-WHEELS.

239. THE paddle-wheel is a circular framework of bars of iron and flat boards or vanes turned by the engine. The vanes dip below the water during a certain part of the time of every revolution, pressing forcibly on the water as they are turned, more forcibly the faster they move, as the only resistances to their motion which exist are those due to friction and the inertia of the water, which will not suddenly assume the velocity of the float. The resistance to motion of the floats forces the vessel onward. This will be readily understood when it is considered that the wheel has a regular motion with respect to the ship. Now, a vane, or float, on the wheel is stopped in its motion by the water, that is, it has very little motion relative to the water, but its motion relative to the ship is unchanged, hence the ship must move relatively to the water.

In producing resistance to the motion of a float, a small motion must be given to the water, and this is less as the floats are made larger, and as they move with greater velocities. Thus the float moves backwards a short distance with respect to the general mass of water, and hence the ship need not move so far forwards as before to maintain its proper motion in relation to the float. The velocity of this motion

backwards of the floats is called the *slip* of the paddle-wheel, and it is evidently equal to the difference between the velocities of the float with respect to the vessel and of the vessel with respect to the water.

240. **Feathering Paddles** have floats which keep a nearly vertical position while immersed. To enable them to do so they are attached to the rim of the wheel by pins, which allow them to change the angles which they make with the radii during immersion.

In Fig. 70, A is a pin attached to the arm B C. An arm like B C is provided for each end of every float, and these arms are attached to rings on the inside boss C. A projecting stem A D from each float is connected by means of a guide-rod D E to the eccentric E F, which is rigidly connected with the wheel. In this way the float gets sufficient motion about A to keep it nearly vertical during immersion. Sometimes the eccentric E F is attached to the ship's side, that is, a pin is placed on the ship's side a little in advance of the wheel, and on this there is a sliding-ring or collar to which the rods are attached. In this case, one of the guide-rods is rigidly attached to the sliding-ring to give it motion.

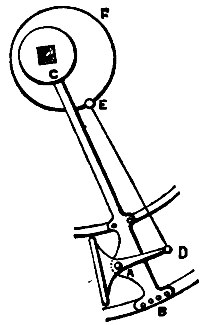

FIG. 70.

Two rings or collars on the boss C have sockets into which the arms B C are riveted. B C is also riveted to two rings, in the way shown in the figure. The inner rings or collars are nearer each other than the outer rings, that the supports for the shaft may be more directly above the centre of gravity of the wheel. The arms have diagonal braces. Sometimes there is

only one ring for each arm, that at B being removed and the arm being curved at its end into a bracket to support A.

If a float is detached in a rough sea, it may be readily replaced by a *fixed float* when two rings have been provided.

Floats of feathering wheels are of pine or elm, $2\frac{1}{2}$ in. to 3 in. thick, formed of two pieces, bolts passing through the whole breadth edgeways. All bolts or pins about the floats must be large to allow for corrosion, and the nuts ought to have square heads, and plates of steel to act as washers. Sometimes the float is of $\frac{3}{4}''$ or $1''$ boiler-plate.

The *area* of a float is investigated in Art. 247. The *length*, measured at right angles to the axis of the vessel, is twice the *breadth* usually. The *pitch*, or distance asunder of the centres of two floats, is usually once and a half to twice the breadth.

The extreme immersion of the lower edge of a float is once and a quarter to twice the breadth. The radius of the circle of the centres of floats is from 2 to $3\frac{1}{2}$ times the breadth of a float. Each float is provided with two short steel journals at its ends, these journals working in lignum-vitæ bushes at A on the arms B C.

For these floats to enter and leave the water edgewise, it will be found that the planes of their faces, if produced, will pass through the highest point of the circle drawn through their centres. Therefore for construction this circle is drawn, also the water-line and three floats, two with their centres at the water-line and one at its lowest position. Draw the stems A D, which may be ·6 of the breadth of a float. Find the centre of the circle which will pass through the three points D, and this is the required centre of the sliding-ring, or collar.

The central boss has a square eye, and is usually keyed to a square shaft with eight keys, the shaft being

burred up against the ends of the keys to prevent their loosening. As the wheels and their shafts are apt to shift their positions relatively to the ship in rough seas, it is found advisable to use large fillets at all the bearings. Bourne recommends bearings formed of continuous curved surfaces, so that the fillets are very large indeed. Steel plates tightened by keys are sometimes employed for the ends of paddle-shafts to prevent this lateral shifting.

The outer bearing of a paddle-wheel shaft is on a *spring-beam*, which rests on the ends of two *paddle-beams*, and these go right across the vessel. The inner bearing is on the frame of the engine.

241. **Paddle-wheels with Radial Floats.**—When a float is always directed to the centre of the wheel, the normal to its surface is parallel to the direction of motion of the vessel only when in its lowest position, and it enters and leaves the water at a very acute angle with the water-surface. Now fluid pressure on a float is normal, so that part of this pressure is so communicated as to raise or lower the vessel.

It is for this reason that radial paddles are never deeply immersed. When the immersion is small the slip is great, so that these floats slip considerably even when made very long.

The usual radius is 2 to $2\frac{1}{2}$ times the greatest immersion of a float, when the vessel is loaded, and one float is usually provided for every foot in diameter of the wheel, so that they are more than 3 feet asunder.

The *area* of a float is given in Art. 248. The breadth is usually 2 to 3 feet. The floats are fastened by means of hook bolts, which enable the engineer to shift their positions (Art. 244).

Very often three sets of arms and rings are provided.

It is known that when a point on the wheel has a velocity of revolution with respect to the ship greater than that of the ship with respect to the water, it will, during a revolution, describe with respect to the water a cycloidal curve called a *curtate* cycloid which has loops indicating slip.

A point on the wheel which has no slip, that is, whose velocity of revolution is the same as the velocity of the ship, describes a cycloidal curve, which has no loops, being an ordinary *cycloid*, and the circle which this point describes round the centre of the wheel has been called the *rolling circle*. Points within this circle describe, with respect to the water, *prolate* cycloids; they move more slowly than the ship, and are seldom allowed to be beneath the surface of the water, as they would drag the water in the direction of the ship's motion.

Every point on the radius outside the rolling circle describes a curtate cycloid with loops. When a number of radii are taken very near each other, and loops drawn for many separate points on each, there is a certain system of loops whose points of intersection are on a straight line. If the points whose loops fulfil this condition be joined by a curved float, it will be found that this will enter and leave the water without producing foam.

To roughly approximate to this condition, floats are formed in two halves, the lower being placed on the after, and the upper on the fore part of the paddle-wheel arm, an expedient which has led to good results.

The common radial wheel is lighter than the feathering paddle, is less easily hurt, and does not need such good workmanship. For these reasons, many people still recommend the radial wheel for large steamers, particularly when these are designed for long voyages or for foreign stations, where workmanship is expensive.

Feathering paddles are much more efficient, however, and are particularly so when there is a variable immersion.

242. **The Centre of Resistance** of a float is a point at which we may suppose a force equal to the reaction of the water to act, at right angles always to the radius of the wheel, such that in one revolution the same work will be done as is really performed by all the floats on the water.

In Art. 241 we referred to a point on the arm which moved backward with respect to the vessel as fast as the vessel moved forward. This point has no motion with respect to the water. If the distance of this point from the centre, that is, the radius of the rolling circle, is r_1, and its velocity with respect to the ship v_1, and if the distance of a point on the float from the centre is r and its velocity v, then $v - v_1$ is the slip at this point. Now, in fluids, resistance to motion varies as the square of the velocity within ordinary limits; hence the resistance at this point $\propto (v - v_1)^2$.

If l is the length of the float (or breadth of the paddle-wheel), and $l\,d\,r$ a small area, the pressure on this area $= k\,l\,(r - r_1)^2$, where k is almost always a constant.

If the water is level and the floats are radial, all the points in a float pass through arcs of concentric circles. When the floats are feathered, the distances moved through by the points are only equal to the chords of these arcs.

It is not necessary for practical purposes to give the results of these calculations. The centre of resistance is always below the middle point, being higher for deep immersions.

243. **Slip.**—If $u =$ velocity of centre (centre of resistance) of float, and $v =$ velocity of vessel, then $u - v$ is the slip which may thus be expressed in nautical miles or knots per hour. It is more commonly expressed as a fraction of the velocity of the float, or

$$\frac{u - v}{u}$$

If the float always enters the water in the same way, the resistance at the float $\propto (u - v)^2$, and resistance to the vessel

$\propto v^2$, hence $(u-v)^2 \propto v^2$, or $u-v \propto v$. Hence, also $u-v \propto u$, or

$$\frac{u-v}{u} = \text{a constant} \quad . \; . \; . \quad (1)$$

That is, the **slip expressed as a fraction** is constant for all speeds if the floats are always immersed in the same way.

It may be shown from this that the efficiency of a feathering paddle is not much affected by change of speed, therefore the assumption of Art. 235 will apply to the case of a vessel propelled by feathering paddles.

244. **Reefing of Paddle-wheels.**—Floats must be capable of being removed to greater or less distances from the centre of the wheel, to suit different draughts; hence they are connected to the arms by hook-bolts, which enable them to be shifted from time to time.

By reefing a wheel we change the relative velocities of the float and of the piston. Let V be the velocity of the piston, and let v the velocity of the ship $= m$ V, then reefing the wheel will decrease m; in fact, m will vary with the distance of the float from the centre of the wheel. If P is the mean pressure on the piston, as P V represents the work done in a unit of time, then by Arts. 233, 235—

$$\mathrm{P\,V} \propto v^3 \text{ or } m\,v^2\,\mathrm{V},\ \mathrm{P} \propto m\,v^2$$

hence

$$v \propto \sqrt{\frac{\mathrm{P}}{m}} \quad . \; . \; . \quad (1)$$

Again, since $\mathrm{P\,V} \propto v^3$ or $m^3\,\mathrm{V}^3$, $\mathrm{P} \propto m^3\,\mathrm{V}^2$

hence

$$\mathrm{V} \propto \sqrt{\frac{\mathrm{P}}{m^3}} \quad . \; . \; . \quad (2)$$

From (1) $m \propto \frac{1}{v^2}$ and from (2) $m \propto \frac{1}{V^{\frac{2}{3}}}$, hence $v^2 \propto V^{\frac{2}{3}}$ or

$$v^3 \propto V \quad . \quad . \quad . \quad (3)$$

From (1) and (2) we see that as m decreases, v and V will increase, but V much more rapidly than v.

Hence, when steam is escaping by the safety-valve of the boiler, and it is therefore necessary to make the engine move faster, reefing the floats will enable us to increase the engine-speed so as to use all the steam. If the floats enter and leave the water in the same way as before, the increased H.P. is effectively used in increasing the speed of the vessel.

From (3), we see that when changes in the velocities of the piston and ship are produced by reefing (not by changes in the steam pressure), the velocity of the piston varies as the cube of the velocity of the vessel.

In considering (1) and (2), it must be remembered that the slip is assumed to be unaltered by reefing, an assumption which is true only when the alteration in position of the floats is small.

245. **Methods of Disconnecting Paddle-wheels.**—Maudslay makes the paddle-box a little wider than usual, so that the wheel, the shaft, and one-half of a crank somewhat like Fig. 37, may be laterally drawn away from the engine. The half-crank on the paddle-shaft is thus drawn from the crank-pin, which is fixed on the half-crank of the engine.

In some tug steamers the intermediate shaft is moved instead of the paddle-wheel shafts.

Instead of the ordinary paddle-crank, Braithwaite used a disc surrounded by a sliding-ring, in which there was a hole to receive the crank-pin. The ring always revolved with the engine, but the paddles could be made to revolve only when a strong key was

inserted between a brass bush on the ring and the disc.

With this excellent arrangement a few taps on the keys will disconnect the paddle-shaft even when the engine is going at full speed.

Grantham protruded a hollow bush from the paddle-shaft crank-eye to receive the crank-pin.

CHAPTER III.

PADDLE-WHEELS (*continued*).

246. ALL instruments employed for the propulsion of vessels act by giving a backward motion to the water, the resistance opposed by the fluid to being moved reacting on the propeller, moving it and the vessel in the opposite direction.

When the vessel's speed is uniform, its resistance to motion is exactly balanced by the reaction of the water on the propeller, whether this propeller is a paddle-wheel with radial floats or one with feathering floats, or a screw, or a jet.

The reaction of the stream of water set in motion by the propeller is measured by the momentum given to it in a unit of time. The mass of water set in motion in a unit of time depends on the speed of the vessel and on the previous velocity of the water acted upon by the propeller. Now in the case of paddle-wheels, we shall not at present consider the motion of the water previous to being acted upon by the propeller.

247. From this we get the following rules for **Feathering Paddles** :—

(1) *The area of the floats.* Find the resistance of

the ship (Arts. 232—3). Divide by the intended speed of the centres (centres of resistance would be more correct) of the floats relative to the water driven back (this is merely the slip); by the intended speed of the same centres relatively to the vessel (this is speed of vessel + slip), and also divide by the mass of a cubic foot of water, to get the area of two floats. With the symbols of Arts. 230 and 243, the **area of a float** is—

$$\frac{\mathrm{K\,A}\,v^2}{m\,u\,(u - v)}$$

where u and v are expressed in knots.

m is the mass of a cubic foot of water = 2 for salt water = 1·94 for fresh water.

(2) The *mean moment of torsion* on each paddle-shaft is half the resistance of the vessel at its greatest intended speed × the radius of the centres of floats (really centres of resistance).

(3) The *efficiency* neglecting friction

$$= \left\{1 - \left(\frac{t}{v - t}\right)^2\right\} \frac{v - t}{u}$$

where t is the forward velocity of the water before being acted on by the paddles.

248. In **Radial Paddles** we have a more complex action. The following rules are the results of investigation by Mr. Napier (Trans. of the Inst. of Civ. Eng. of Scot., 1863–4):—

(1) In finding the *area of floats* proceed as for feathering paddles, letting u be the velocity of the *outer edge* of the float, and having regard to the varying immersion of vessels.

(2) The mean *moment of torsion* on each paddle-

shaft = $\frac{1}{2}$ resistance of vessel × outer radius of wheel ×

$$\sqrt{\frac{\text{outer radius of wheel}}{\text{height of centre of wheel above water}}}$$

(3) The *efficiency* = efficiency of corresponding feathering paddle ×

$$\sqrt{\frac{\text{height of centre of wheel above water}}{\text{outer radius of wheel}}}$$

CHAPTER IV.

THE SCREW-PROPELLER.

249. FIG. 71 shows the ordinary form of double-bladed, and hence double-threaded, screw applied to vessels. It is keyed very firmly (Art. 255) to a shaft at M D. As it revolves it forces its way through the water as an ordinary screw does in its nut, or rather, as a large wood screw with wide threads would force its way through a piece of dough. Its rate of revolution would force the screw forward in its solid nut much more rapidly than it does in the water, because it is only when the velocity of water normal to a surface *differs* from the velocity of the surface in the same direction, that resistance is experienced, and this resistance in the case of a propeller must be very great.

250. Let A B, Fig. 72, be a vertical line which never changes its position. Let A C be a line always horizontal and passing through a point on A B, but changing its position in such a way that one extremity revolves regularly round A B as an axis, and the other extremity travels regularly along A B. Evidently C is always on

the surface of a cylinder, and when such lines as A C and B D are at equal vertical distances asunder, points such as C and D will be at equal horizontal distances asunder. It is evident that the path of every point in A C, for instance that of C^1, is a curve.

A B is called the **length** of the screw, being the dimension of the blade taken *parallel to the axis.* If the portion of the line is continued until D comes vertically under C again, the distance which A has travelled is called the **pitch** of the screw. We see that *the pitch is the distance between two threads measured in a direction parallel to the axis.*

The pitch is often as much as 3c feet, so that only a small part of each thread is shown, such as would be obtained by cutting a spiral stair by two horizontal cuts very near each other.

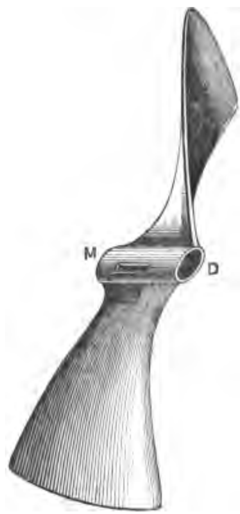

FIG. 71.

C D E is the **angle** of the screw. It is the angle between the curve C D at D and the plane B D E which is at right angles to the axis A B. It is evident that the angle $C^1 D^1 E^1$ is greater than C D E, so that *for the same pitch the angle of a screw increases as the radius decreases.*

Sometimes the angle of the screw changes in value, that is, the curve C D is not always inclined at the same angle to the horizontal, but gets steeper towards D, let us say. Then the

pitch of the screw-thread increases towards D. In such a case as this, C is the *advancing edge* which may really have no slip, and to whose motion, therefore, the water offers no resistance. D is the *following edge*, which has a great amount of slip.

Referring to Fig. 72, we may illustrate the above change in the shape of the blade, by supposing the surface A B D C to be firmly held at the parts A C^1 D B and by C being pushed backward so as to make the bend in the plate less rapid at that place.

Holm's screw, which is now disused, was defective through this change in the pitch being too great, so great indeed was it, that at D the edge of the blade was almost parallel with the axis.

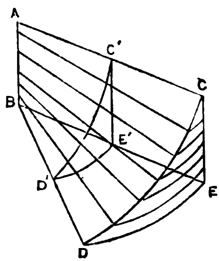

FIG. 72.

The water always gets a motion normal to the surface of the screw, and a particle tends to get away from the axis as well as to move backwards. Screws of great pitch and small radius, that is, screws whose angles are great, exert much outward pressure, moving the water with what has been called a centrifugal force. This outward motion tends to raise a wave behind the vessel, which is found useful in head-winds, but which, as a rule, is regarded as an indication of a waste of energy.

Now, the central parts of all ordinary screws produce this effect, and hence it is often found convenient *to decrease the pitch towards the axis.*

If the line A B (Fig. 72) be made less than C E, so that in constructing the screw C has a greater vertical velocity than A, we get a screw whose pitch decreases towards the axis. Perhaps the change is better indicated by supposing D C held firmly, whilst A B is made a curved line convex to the paper.

The screws of the Earl of Dundonald and of Mr. Hodgson tended by the configuration of the blades to make the particles of water take a motion which was almost altogether in a

backward direction. Holm's screw, already spoken of, also to a certain extent prevented outward motion of the water.

251. When a screw works in a solid nut it moves forward by a distance equal to the pitch during every revolution. In consequence of the yielding nature of the water, it is from the resistance it experiences in giving the fluid a certain velocity backwards that the screw is enabled to force the vessel onwards. The difference between the rates at which a screw would force its way through an unyielding nut (this is always called the velocity of the screw) and through water is called the **slip.** The slip is often expressed as a *fraction of the velocity of the screw*, and it may also be expressed in nautical rules or *knots per hour.*

If v is the velocity of the vessel in feet, n the number of revolutions of the screw in a unit of time, p the pitch of the screw in feet—

$$\frac{np - v}{np} = \text{slip expressed as a fraction.}$$

In practice the slip varies from 0·1 to 0·3.

There is *negative slip* when np is no longer greater than the velocity of the vessel. When no sails are used this slip answers from the screw working in the current of water formed by friction on the vessel's sides. It is not to be supposed that there is an increase of efficiency in such a case.

When through the use of its sails a screw has much negative slip, the engineer ought to observe narrowly the thrust on the shaft. If this ceases, the screw ought to be disconnected, as it is now *dragging.*

The *diameter* of the screw is twice the line A C, being the diameter of the cylinder on which the screw-line C D is traced.

The *area* of the screw is half the area of the *total* surface of the blades.

The *thrust* of the screw may be measured by a dynamometer (Art. 150). It is equal to the resistance to motion of the vessel. The resistance, or thrust, multiplied by v is the work done in moving the vessel in a unit of time. When this work is compared with the work given out by the engine in a unit of time, we get the *efficiency* of the propeller. The efficiency is found to vary from ·66 to ·55 in screw-propellers.

The screw is placed in front of the rudder in a large rectangular hole in the dead wood, the shaft from M D (Fig. 71) passing through the stern to the engine. This shaft revolves at a rate varying from 40 to 200 revolutions per minute, and is worked either from the engine directly, or by means of gearing.

When it is possible to employ screws of large diameter, and to immerse them deeply, the ordinary two-bladed propeller (Fig. 71) is most commonly employed and is perhaps most efficient. But if the diameter is restricted, and if the vessel is required for towing, and if it has often to encounter headwinds, more than two blades will often be found necessary.

Screws of two blades can be effectually disconnected, as they may be lifted out of the water; hence they are commonly used where sails are much employed and disconnection is often required for great length of time, as in ships of war. It will be seen from experiments on the *Pelican* (Art. 270) that a two-bladed screw of constant pitch is almost as efficient as any other.

Every time that a blade passes the stern-post a blow is given to the vessel by the moving water, producing a horizontal vibration of the stern. When an even number of blades are used these vibrations partially destroy each other.

252. **Griffith's Screw-Propeller.** — There is much friction at the centre of the arms and at the boss of an ordinary screw. Griffith introduced a hollow sphere on whose outer surface three blades were placed in proper positions. Those vibrations which accompany the motion of screws, due to the great angle of the inner parts of the blade, are here much lessened. To reduce these still further four blades might be employed. The sphere has a diameter which is one-third of

that of the screw; each blade at the root has a breadth equal to the diameter of the sphere, tapering to two-thirds of this at the circumference. This construction was adopted to allow for the varying velocities through the water of different parts. It is questionable whether it has any advantage besides giving increased steadiness. When it is desirable to disconnect the screw, the moveable blades are turned round so as to be parallel to the direction of motion of the ship, in which position they offer little resistance forward.

The pitch may be increased, if necessary, when sailing under canvas; but this never leads to much increase in efficiency, as the blade forms part of a screw at one certain pitch and at no other. A blade when broken may be readily replaced, weighing as it does only one-sixth of the weight of the whole screw.

Beattie's Screw works outside the rudder, which is divided, the shaft passing through the rudder-stock. The vibrations which usually accompany the motion of screws are here much lessened.

Maudslay's Feathering Screw has arms which, like those of Griffith's propeller, may separately be brought parallel to the direction of the ship's motion, and held in that position.

Ericson's Propeller has a number of blades fixed at a distance from the axis on a ring, which is connected to a central boss by two or more arms.

253. **Twin Screws.**—Two screws are useful in giving a considerable propelling area when there is little draught; since they are worked independently they enable the vessel to be steered if the rudder is defective or when rapid evolution is necessary, and one screw may be used alone for propulsion when the other is damaged. They are easily hurt, they complicate the machinery, and by the increased number of bearings outside and inside add greatly to the wear and tear. They are, perhaps, most usefully employed in river-steamers of small draught required to move in tortuous channels.

A screw A may be replaced by twin screws B B when the dimensions of B = dimensions of A $\div \sqrt{2}$, and when the number of revolutions per minute of B = number of revolutions of A $\times \sqrt{2}$. It will be found that the mean amount of

torsion on each of the two shafts is the mean torsion of A's shaft $\div 2\sqrt{2}$.

254. **Jet Propulsion.**—"The *Waterwitch*."—The *Waterwitch* is iron plated, is 162 feet long, and is propelled by means of two jets of water flowing out at the sides in the direction of the stern, just beneath the water surface.

A large centrifugal pump brings the water from beneath the ship. The vessel moves as a rocket moves upwards, as Hero's engine moved and as the water-wheel is moved in Barker's mill.

The investigations regarding feathering paddles and jets are simpler than those of other propellers, as the sections of the streams of water driven astern are simply the areas of the float and of the jet-nozzle. If a is the area of the nozzle, u the velocity of the jet of water in feet with respect to the orifice, and m is the mass of one cubic foot of water, then the whole propelling force is $m\,a\,u\,(u - v)$, where v is the velocity of the ship. This is equal to the resistance of the vessel, therefore the work done on the vessel in a unit of time $= m\,a\,u\,(u - v)\,v$. Now, the kinetic energy given to the stream in a unit of time is $\frac{1}{2}\,m\,a\,u\,(u - v)^2$, so that the *efficiency* of the propeller, neglecting friction, may be shown to be—

$$\frac{m\,a\,u\,v\,(u-v)}{m\,a\,u\,v\,(u-v) + \frac{1}{2}\,m\,a\,u(u-v)^2}, \text{ or } \frac{2v}{u+v}.$$

This efficiency becomes more and more nearly unity as u approaches v in value—that is, as the water is more and more nearly motionless leaving the orifices. Now, it may be shown that u can never be as small as v; also, as $u - v$ becomes smaller, the area of the orifice must increase for the same ship; hence we find that *the greater the quantity of water operated upon by a jet propeller, the greater is the efficiency.*

Only for friction in the pipes the jet propeller acting in still water would be a propeller of maximum efficiency, but

as much as 35 per cent. of the indicated H.P. of the engine is expended in friction in the pipes.

There are objections to increasing the size or number of the orifices; there are objections to the use of any orifices, one of which is that they increase the resistance of the vessel; and it seems that the jet as at present employed is much less efficient than other propellers, unless in special cases such as that of a gun-boat requiring quick evolution.

255. **Screw-Shaft.**—The screw is attached to the shaft by means of deep keys. Of these keys there are often two driven in at the after side of the boss to meet others already fitted to the shaft. The edges of the key-beds are burred

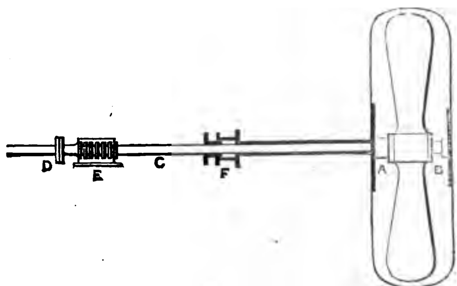

FIG. 73.

over the keys to prevent their working loose. A bolt also is tapped into the shaft, passing through the boss of the screw.

The shaft A B (Fig. 73) supporting the screw, is a brass pipe, closed at B, and resting in a bearing. At the end A, the pipe is square in bore, so that it may receive the wrought-iron screw-shaft A D.

A disc at the end B in the bearing receives the backward thrust of the screw.

The screw-shaft A D passes through a long brass pipe A F, placed as a bearing in the dead wood. Sometimes the bearing is lined with lignum vitæ, and then the shaft is covered with brass, as brass and wood work well together.

There is a stuffing-box at F to prevent entrance of water which has free access to the bearing itself. A water-pipe, supplying water to keep all the bearings cool, is carried above the shaft along its whole length.

E shows the usual thrust bearing. The shaft has many collars at that place, these collars rest in grooves which have been turned out of the solid iron, and are lined with a soft metal and receive the thrust of the shaft.

The shaft to the engine is made in different lengths, provided with bolted flange couplings, such as are now getting common in millwork.

The mean moment of torsion on the shaft = thrust of screw × pitch × 6·2832).* The thrust of the screw may be supposed to be about 3 per cent. more than the resistance of the vessel. In Rankine's *Shipbuilding* will be found an investigation of the greatest moments of torsion on various shafts, the motive power being that of one, two, or three engines.

256. In ordinary steam-boats, which depend almost altogether on their engines for propulsion, it is seldom necessary to **disconnect** the propeller. Hence no means are ever provided in these vessels for lifting the screw, and whenever the engine breaks down the screw is allowed to revolve freely as the vessel proceeds under canvas. Near the stern, instead of an ordinary coupling, discs are keyed on the ends of the shaft on which the screw is and of that which proceeds to the engine, and from one of these discs a number of pins are screwed forward so as to pass through holes in the other.

In the royal navy arrangements are made when necessary for withdrawing the screw-shaft from the brass shaft supporting the propeller, thus enabling the iron frame supporting the screw to be lifted. This is done by means of screws, a rack and worm, or by means of ropes and pulleys.

257. The **pitch of the screw** is evidently equal to $2\pi AC \times$ tangent of the angle of the screw. That is, if p = pitch in feet R = radius in feet, and α = angle of screw, then—

$$p = 2\pi R \tan \alpha \quad . \quad . \quad . \quad (1)$$

* Rankine.

258. **Angle.**—When not given the angle of the screw, it is easy to find it by measuring the edge of the propeller C D with a curved rule, or by means of a tape-line; this is the hypotheneuse of the right-angled triangle C E D, which may be drawn with the help of C E or A B, the length of the screw. Since the sine of the angle of the screw $= \dfrac{\text{length}}{\text{C D}}$ we may use mathematical tables in finding this angle.

259. **Area of Surface of Blade.**—This involves an integration, but an approximate result may be found by employing Guldinus's theorem.

The area of one side of a screw-blade is nearly equal to the line A C × the path described normally to A C by its middle point. Now, it may be shown that C D E, $C^1 D^1 E^1$ are curved right-angled triangles; hence, make a right-angled triangle whose height is C E, the length of the screw-blade, and one of whose acute angles is the angle of the screw, C D E; in fact, the triangle referred to above (Art. 258). Bisect the base E D and join with the vertex C. This joining line will be equal to $C^1 D^1$, the path of the centre of the line A C. Multiply this by A C for the area of the screw.

Now, $D^1 C^1 = \sqrt{l^2 + \dfrac{l^2 \pi^2 R^2}{p^2}}$ where l is the length of the screw in feet, hence

$$\text{the area} = R\,l \sqrt{1 + \frac{\pi^2 R^2}{p^2}}$$

260. Relations between the pitch p in feet, the speed of the ship in knots per hour v, the number of revolutions of the propeller n, and the slip expressed as a fraction s.

When the slip is expressed in knots per hour, let it be s^1, then, instead of s in our calculations, we must use $\dfrac{s^1}{v + s^1}$

Since there are 6,080 feet in a nautical mile, it is evident that

$$\frac{60\,p\,n}{6080}(1 - s) = v, \text{ which becomes}$$

$$p\,n\,(1 - s) = \frac{304}{3}v \quad . \; . \; . \; (3)$$

261. (*See* Arts 246—7.) The screw gives diverging motions to the water, and the slip of the water is less than the slip of the screw in the proportion of the square of the cosine of the angle made by the screw-surface at the given place, with a thwart-ship plane. This angle is different at different points on the propeller. By the method sketched in Art. 246, we may find πr^2 (the cross-section of the cylinder of water acted upon, where r is the *radius* of the screw. It is necessary first to find the area of two floats of the equivalent feathering paddle. Rankine by another method finds the area of the floats equal to—

$$\pi r^2 \left(1 - \frac{\cdot 127\, p}{r}\right)$$

where p is the pitch of the screw. So that r may be found.

The Efficiency in still water neglecting friction

$$= 1 - \frac{\text{real-slip of propeller}}{\text{speed of propeller}}$$

This includes the loss from rotatory and other motions given to the water.

In water already disturbed there is a loss proportional to the slip together with a loss proportional to the square of the velocity of water previous to being acted upon by the screw.

Before proceeding, we may notice that the screw does not fulfil the conditions required in Art. 235, so that the locomotive performance $D^{\frac{2}{3}} v^3$ is not so nearly constant in ships of the same shape for similar screws as for similar paddles. Experiment shows that the variation is small, and it is found convenient to employ the same methods for making calculations in both cases, and for these methods students will refer to Arts. 235—7.

In determining the speed most economical for a screw-vessel against a current (*see* Art. 236), it must be remembered that the velocity of the ship due to the coals used is much less in this case than when there is no current.

262. We proceed to the calculation of the thrust of the screw from its shape, leaving friction out of account.

Let $C^1 D^1$ (Fig. 72) be any one of the curves generated by a point on AC. Let the angle $C^1 D^1 E^1 = \theta$, and let $AC^1 = \rho$, $AC = r$, CDE the angle of the screw $= \alpha$.

Let the screw make n revolutions per minute, then the velocity of a point in the line $C^1 D^1$ in a direction normal to the surface is $2 \pi n \rho \sin \theta$. The velocity of the same point normal to the surface due to the motion of the vessel $= v \cos \theta$ if v is the velocity of the ship; hence, the velocity of the point with respect to the water is $2 \pi n \rho \sin \theta - v \cos \theta$. But resistance to motion in a fluid varies as the square of the velocity; hence the pressure on the unit area along the line $C^1 D^1$ is $K_1 (2 \pi n \rho \sin \theta - v \cos \theta)^2$, where K_1 is a constant at moderate speeds.

Resolving this pressure along and at right angles to the axis of the screw, we get—

$$K_1 (2 \pi n \rho \sin \theta - v \cos \theta)^2 \cos \theta, \text{ and}$$
$$K_1 (2 \pi n \rho \sin \theta - v \cos \theta)^2 \sin \theta.$$

Now, the pressure in the direction of the axis is equal to the resistance to motion of the vessel, or $K A v^2$, where A is the area of immersed midship section, and where K has been determined as before by experiment and expresses resistance per unit area. Hence—

$$K A v^2 = 2 K_1 \pi n \int_0^r \cos \theta \,(2 \pi n \rho \sin \theta - \cos \theta)^2 c \, dr,$$

where c is the length of the line $C^1 D^1$. Now as p is the pitch,

$$\frac{p}{2 \pi \rho} = \tan \theta, \text{ hence}$$

$$\sin \theta = \frac{p}{\sqrt{4 \pi^2 \rho^2 + p^2}}, \text{ and}$$

$$\cos \theta = \frac{2 \pi \rho}{\sqrt{4 \pi^2 \rho^2 + p^2}}, \text{ and}$$

$$c = \frac{2 \pi l \rho}{p \cos \theta} = \frac{l}{p} \sqrt{4 \pi^2 \rho^2 + p^2}$$

When these values are introduced into the above equation, and when it is simplified, we get

$$K A v^2 = 16 \pi 4 K_1 n (n p - v)^2 \frac{l}{p} \int_0^r \frac{\rho^3 \, dr}{4 \pi^2 \rho^2 + p^2}$$

The integral is found to be $\frac{r}{2}$ $(1 + \tan^2 \alpha \log \sin^2 \alpha)$, and instead of np we may write u, the velocity of the screw in advance, and for $\frac{u - v}{u}$ we may write s the slip; so that we get eventually—

$$s = \frac{1}{1 + \sqrt{\pi \frac{l}{p} \cdot \frac{r^2}{A} (1 + \tan^2 \alpha \log \sin^2 \alpha)}}$$

From this we get the rules :—

(1.) *The slip is diminished by increasing the angle of the screw.*

(2.) *The slip is diminished by increasing the radius.*

(3.) *The slip is diminished by increasing the length.* (It will be noticed that, since much friction is introduced by increasing the length, diminution of slip in this last case may not be attended with any increase in efficiency.)

When, in the above formula, we let $\frac{p}{2r}$, the ratio of the pitch to the diameter of the screw, be called e, and substitute for $\tan \alpha$ its value $\frac{p}{2\pi r}$, we get

$$s = \frac{1}{1 + \sqrt{\frac{l}{p} \pi \frac{r^2}{A} \left(1 + \frac{e^2}{\pi} \log \frac{e^2}{\pi^2 + e^2}\right)}}$$

or

$$\left(\frac{s - 1}{s}\right)^2 = \frac{l}{p} \cdot \pi \cdot \frac{r^2}{A} \cdot \left(1 + \frac{e^2}{\pi^2} \log \frac{e^2}{\pi^2 + e^2}\right)$$

a formula which is approximated to by the results of experiments on the *Pelican* steam-ship, Chapter V.

Taking the pressure at right angles to the axis of the screw and proceeding as before, we find the work done in one unit

of time in turning the screw, or $2\pi K_1 n \int \rho \sin\theta (2\pi n \rho \sin\theta - v\cos\theta)^2 c\,d\,r$ to be $4 k \pi^3 l n (n p - v)^2 r^2 (1 + \tan^2 \alpha \log_e \sin^2 \alpha)$; that is, if P is the pressure on the piston of the steam-engine, and if V is its velocity—

$$P V = 4 K_1 \pi^3 l n (n p - v)^3 r^2 (1 + \tan^2 \alpha \log_e \sin^2 \alpha).$$

which becomes when simplified—

$$P V = 2\pi K_1 . \frac{l}{p} . s^2 . v^3 . r^2 (1 + \tan^2 \alpha \log_e \sin^2 \alpha)$$

From this we have our former result, for since $P V \propto$ H.P., the H.P. $\propto v^3$ if K_1, s, l, p, and α remain constant.

263. As friction and other considerations have been neglected, the above investigation is only of use in helping us in experiments. In the next chapter will be given the proportions proper for all screws, as obtained from experiments on the *Pelican* steam-ship by MM. Bourgois and Moll.

CHAPTER V.

PROPORTIONS AND EFFICIENCY OF SCREWS.

(*As determined from experiments by MM. Bourgois and Moll on the steam-packet* Pelican *with screws of two, four, and six blades. Special attention was directed to screws of four blades.*)

264. IN the following determinations A = area of immersed midship section; v = speed of the vessel in knots per hour; R = radius of the screw in feet; K is the resistance in pounds per square foot of the immersed midship section, a

quantity which is nearly constant in value, for all speeds (*see below*); p = pitch of screw (or average pitch when it is variable). Hence $\frac{p}{2R}$ expresses the ratio of the pitch to the diameter. Let $\frac{p}{2R}$ be called e, l = length of screw. Hence $\frac{l}{p}$ will express the length as a fraction of the whole pitch.

s = slip expressed as a fraction of the velocity of the screw.

$$K \cdot \frac{A}{4R^2} \text{ or } K \times \frac{\text{area in square feet of immersed midship section}}{\text{square of diameter of screw in feet}}$$

bears a constant relation to the relative resistances of the hull and screw, and is technically called the *Relative Resistance.* This definition must be well remembered.

On the *Pelican*, A and therefore the relative resistance, was modified by means of a moveable plane, lowered into the water at the bows, and it was found that increasing the area of the hull was equivalent to proportionally decreasing $4R^2$ in its effect on the pitch, length, angle, and efficiency of the screw.

265. Now, since most of the experiments were made at a speed of 9·4 knots per hour, we may suppose the work done in one hour to be equal to—

$$K A v^3, \text{ or } K A (1 - s)^3 p^3 n^3.$$

where n is the number of revolutions made in one hour.

A may be measured, K determined by separate experiments and found to be constant at *moderate speeds*, but to be equal to $11 \cdot 9097\, v^{0 \cdot 28}$, when the speed became very great (see also Rankine's method for determining K A, Art. 230). Hence, in the above ex-

pression s and p and n have to be determined. In these determinations the effects of winds and variable draught were experimented upon and eliminated.

266. **How the Slip is affected by the Speed.**—The normal speed was 9·5 knots, for any other speed the $slip = slip\ at\ 9{\cdot}5\ knots \times \left(1{\cdot}098 - \frac{v}{100}\right)$ where v is the new speed.

The experiments on the *Pelican* may be repeated on other well-shaped vessels, by supposing the speeds at which the same effects are produced to vary as the square roots of the linear dimensions of the vessels.* Thus, in a vessel larger than the *Pelican* the normal speed would be greater than 9·5 knots. This is evident, for s is *very nearly* the same in all similar vessels provided with similar screws (see end of Art. 261).

267. **How the slip is affected by the length, or rather by $\frac{l}{p}$.**

For a screw whose pitch was 14 feet—

$$Slip\left(when\ \frac{l}{p} = {\cdot}75\right) - slip\left(when\ \frac{l}{p} = .30\right) = {\cdot}05$$
$$or\ {\cdot}06$$

And the changes in slip were greater when $\frac{l}{p}$ was small, the slip being nearly constant in value on both sides of $\frac{l}{p} = {\cdot}75$.

For screws of less or greater pitches than 14 feet the difference above mentioned is less than ·05.

268. **How the Slip is affected by the length and by the Pitch.**—The slip is found to increase in an

* Scott Russell's rule, Art. 229.

arithmetical progression when the pitch increases in a geometrical progression. The following results may be depended upon for *four-bladed screws* :—

$$\text{When } \frac{l}{p} = \cdot 75,\ s = \cdot 1580 + \cdot 555 \log e.$$
$$\text{When } \frac{l}{p} = \cdot 6,\ s = \cdot 1720 + \cdot 555 \log e.$$
$$\text{When } \frac{l}{p} = \cdot 45,\ s = \cdot 1858 + \cdot 555 \log e.$$
$$\text{When } \frac{l}{p} = \cdot 375,\ s = \cdot 2000 + \cdot 555 \log e.$$
$$\text{When } \frac{l}{p} = \cdot 3,\ s = \cdot 2140 + \cdot 555 \log e.$$

For *two-bladed screws* the experiments were not so numerous, but they indicated the above law, that s increased in an arithmetical progression as e increased in geometrical progression.

$$\text{When } \frac{l}{p} = \cdot 45,\ s = \cdot 0718 + \cdot 0566 \log e.$$
$$\text{When } \frac{l}{p} = \cdot 30,\ s = \cdot 0800 + \cdot 0566 \log e.$$

269. **How the Slip is affected by the "Relative Resistance,"**

$$\frac{K\ A}{4\ R^2}$$

It is found that *equal differences in slip are produced by equal differences in the relative resistances*, or in $\frac{A}{4\ R^2}$, that is—

$$s = \cdot 178 + \cdot 0693 \frac{A}{4\ R^2}$$

For all kinds of screws of the diameter 5·51 feet we have the rule—

$$s = \frac{e^{1 \cdot 15 \left(\frac{A}{4 R^2}\right)^{0 \cdot 85}}}{e^{1 \cdot 15 \left(\frac{A}{4 R^2}\right)^{0 \cdot 85}} + c}$$

where c varies with the character of the screw, and is given in the following table :—

	$\frac{l}{p}$	c
Two-bladed	·300	9·743
	·450	11·007
Four-bladed	·300	10·714
	·375	11·600
	·450	12·280
	·600	13·027
	·750	13·453
Six-bladed	·600	13·042

Now when we have to consider a screw of other diameter than 5·51 feet, we merely use instead of A in the above formula the expression $\frac{4 R^2}{(5 \cdot 51)^2 A}$. This is the hypothetical area of midship section, which would have the same relative resistance to a screw of 5·51

feet that the real section has to the real screw. In fact, the equation for screws generally becomes—

$$s = \frac{e^{1\cdot 15\left(\frac{5\cdot 51\ A^{\frac{1}{2}}}{4\ R^2}\right)^{1\cdot 7}}}{e^{1\cdot 15\left(\frac{4\ R^2}{5\cdot 51\ A^{\frac{1}{2}}}\right)^{1\cdot 7}} + c}$$

270. **Efficiency.**—The work transmitted to the screw is evidently the indicated H.P., multiplied by the *efficiency of the mechanism.* This efficiency was calculated for different speeds in our present case from separate experiments. Now, the work done in a unit of time calculated from indications of the dynamometer will of course $= \mathrm{K\,A}\,(1 - s)^3\,p^3\,n^3$ (Art. 265). Hence, the efficiency of the screw

$$= \frac{\mathrm{K\,A}\,(1 - s)^3\,p^3\,n^3}{\text{Indicated H.P.} \times \text{Efficiency of Mechanism}}$$

This efficiency may readily be calculated at any time given the steam *indications*, the speeds of the engine and screw, and the slip.

By a method which differs from that given above, MM. Bourgois and Moll determined the relative efficiencies of different screws. With the largest screws, that is, when the *relative resistances* are as small as possible, $\frac{2}{3}$rds of the work given out by the engine may be expended usefully in producing motion of the vessel. With the greatest relative resistances commonly employed, only ·55 of the work is usefully expended. Hence, in ordinary cases, the efficiency of the screw-propeller in a well-formed vessel varies from—

·66 to ·55.

It was found that the efficiency diminished slightly when the speed became very great. This arose from K being variable at high speeds.

270*a*. After drawing many curves, connecting efficiency and pitch and length, it was found that in the *Pelican* at the speed of 9·5 knots, with propellers whose diameters were 5·51 feet, the following values of e and $\frac{l}{p}$ gave Maximum Efficiencies :—

	$\frac{p}{2R}$ or e	$\frac{l}{p}$	Maximum Efficiencies.
Screws of Four Blades	1·035	0·300	K × ·07495
	1·743	0·375	K × ·07485
	1·842	0·450	K × ·07465
	1·985	0·600	K × ·07405
	2·067	0·750	K × ·07350
Screws of Two Blades	1·520	0·300	K × ·07500
	1·689	0·450	K × ·07690

Screws of two and four blades seem equally efficient in still water, the two-bladed screw having the shorter pitch.

The following table gives the proportions of screws which give maximum efficiencies, and shows that with short pitches or long pitches for the same vessel at the same speed it is possible to make screws of such proper proportions as will give almost the same efficiencies in the several cases.

The diameter of the screw experimented upon was 5·51 feet in every case.

		$\frac{p}{2R}$	$\frac{l}{p}$	s.
Four-bladed Screws.	1st Series, short pitches . . .	1·097	·300	·2351
		1·229	·375	·2485
		1·427	·450	·2710
		1·628	·600	·2865
		1·779	·750	·2960
	2nd Series, long pitches . . .	2·024	·300	·3885
		2·097	·375	·3810
		2·158	·450	·3735
		2·229	·600	·3630
		2·259	·750	·3550
Two-bladed Screws.	1st Series, short pitches . . .	1·283	·300	·2905
		1·173	·450	·2445
	2nd Series, long pitches . . .	1·664	·300	·3575
		2·045	·450	·3825

With $\frac{l}{p}$ and $\frac{A}{4R^2}$ *constant*, $\frac{p}{2R}$ ought to increase when we pass from two to four bladed screws.

270*b*. When the *relative resistance* $\frac{KA}{4R^2}$ is constant, the efficiency increases as the radius or diameter of the screw is increased.

From results of experiments on screws of other diameters than 5·51 feet, we find that, if the practical efficiency of the screw of 5·51 feet is fixed at K × ·0727, that of a screw of 6·72 feet is K × ·0770, and that of

a screw of 8·2 feet is K × ·0820, and each screw will have two pitches which will give the same efficiency of three per cent. less than the *greatest* maximum performance, namely (in the four-bladed screw, when—

$$\frac{p}{2\,R} = 2{\cdot}091, \frac{l}{p} = {\cdot}300\,; \text{ and } \frac{p}{2\,R} = 2{\cdot}552, \frac{l}{p} = {\cdot}450$$

Perhaps the greatest maximum performance for this diameter is reached by a screw whose proportions are intermediate between those of these two.

With modified screws, the pitch increasing along the periphery, or towards the axis, or in both directions (*see* Art. 250), there may be a possible addition of 5 per cent. to the above efficiencies.

When $\frac{K\,A}{4\,R^2}$, the *relative resistance*, varies, changes must be made in $\frac{p}{2\,R}$ and $\frac{l}{p}$ to obtain maximum performances.

270*c*. The following Table for four-bladed screws gives the *greatest* maximum performances attainable in the case of the *Pelican*, and in all cases in which $\frac{K\,A}{4\,R^2}$ is the same as in the *Pelican* :—

$\frac{p}{2\,R}$	$\frac{l}{p}$	$\frac{K\,A}{4\,R^2}$	Highest Maximum Performances.
1·354	0·450	K × 5·425	0·06260 × K
1·683	0·360	K × 3·615	0·07500 × K
1·950	0·310	K × 2·429	0·07950 × K
2·200	0·281	K × 1·632	0·08460 × K

270*d*. The following Table gives the proper proportions of screw-propellers for vessels of different kinds:—

Class of Vessel.	Relative Resistance $\frac{KA}{4R^2}$	Screws of Two Blades.		Four Blades.		Six Blades.	
		$\frac{p}{2R}$	$\frac{l}{p}$	$\frac{p}{2R}$	$\frac{l}{p}$	$\frac{p}{2R}$	$\frac{l}{p}$
... ...	K × 5·5	1·006	0·454	1·342	0·454	1·677	0·794
... ...	K × 5·0	1·069	0·428	1·425	0·428	1·771	0·749
... ...	K × 4·5	1·135	0·402	1·513	0·402	1·891	0·703
Auxiliary Line of Battle-ship . .	K × 4·0	1·205	0·378	1·607	0·378	2·009	0·661
Auxiliary Frigate	K × 3·5	1·279	0·355	1·705	0·355	2·131	0·621
High-speed Line of Battle-ship .	K × 3·0	1·357	0·334	1·810	0·334	2·262	0·585
High-speed Frigate	K × 2·5	1·450	0·313	1·933	0·313	2·416	0·548
High-speed Corvette . . .	K × 2·0	1·560	0·294	2·080	0·294	2·600	0·515
Despatch Boat	K × 1·5	1·682	0·275	2·243	0·275	2·804	0·481

Example.

Required a two-bladed screw—given the greatest immersed sectional area A of a vessel to be 300 square feet, and that the draught will only allow the screw's diameter to be 10 feet, then $\frac{A}{4 R^2} = 3$. By the above table $\frac{p}{2 R} = 1{\cdot}357$; hence, as $2 R = 10$, p must be 13·57 feet.

By the table $\frac{l}{p} = 0{\cdot}334 \therefore l = 13{\cdot}57 \times 0{\cdot}334 = 4{\cdot}53$ feet.

Thus we have diam. = 10, pitch = 13·57 and length = 4·53 feet.

CHAPTER VI.

MARINE ENGINES.

271. ALLEN, Gensanne, Hulls, Bernouilli, Gautier, Genevois, Comte d'Auxiron, Perier, Fitch, Henry, Rumsey, Miller, Cartwright, Earl Stanhope, Livingston, Evans, and De Blancs treated of steam-navigation in writings and by practical experiments during the eighteenth century. Symington, in 1802, built a steamboat which towed two vessels on the Clyde against a strong head-wind. Fulton, in 1803, experimented with a steamboat on the Hudson; but it was not till 1807, when Fulton was provided with engines specially designed for the *Clermont* by Boulton and Watt that the subject became of commercial importance. Bell's boat, the

Comet, on the Clyde, was the first European steam-vessel which was really successful.

American engineers directed their attention to *river* and English engineers to *ocean* navigation. From 1812 to 1837 steam-packets were established between England and ports in Ireland and France, and, eventually, ports on the Mediterranean. In 1836, it was proposed to establish a line of Atlantic steamers. The first project failed of success, and the steamers were withdrawn from the service after a time. A second project, started by Mr. Samuel Cunard, was successful, the company having a government grant of money for mail-service. Regular steam communication is now established between England and the most distant places. A history of the improvements in marine-engineering which enable long voyages to be made would be out of place in this little treatise.

272. In no department of engineering is there so much necessity for the exercise of judgment as in the designing of marine engines. For short trips it may not be necessary to provide the expensive apparatus for obtaining economy which is absolutely necessary when a vessel carries her own coal on a long voyage. Again, the character of the trade the available space, the relation between first cost and current expenses and economy of fuel and other considerations must be attended to in designing engines for steam-packets. In ships of war, the nature of the armament, and expected service, and many other matters affect the character of the engine.

273. Marine engines are becoming more and more efficient, their economy of fuel usually exceeding that of Cornish engines ; this arises from the extending use of superheating and high pressures and speeds (Arts. 130, 165), and, as a consequence, improvements in expansion, the use of steam-jackets, the balancing of

pressures on the crank-shaft to reduce friction, the use of surface condensation (Art. 98), of feed-heaters, and of coverings for boilers and steam-pipes.

274. *Expansion, Steam-jackets, Friction at the crank-shaft bearings.*—In Arts. 61—3, we considered the benefits arising from the use of expansion in steam-cylinders. We have seen that when saturated steam expands doing work condensation occurs. From the laws of thermodynamics it is easy to show that condensation would produce no diminution in efficiency in a well-covered cylinder, were it not that some condensed vapour remains in the cylinder at the end of a stroke, and that the cylinder, consisting as it does of a mass of conducting material, is heated by the condensation of vapour during expansion, and is cooled by evaporation during exhaustion (it will be remembered that condensed vapour is a good radiator and absorber of heat), and thus heat is conveyed from a hot body a cold one without conversion into work (*see* Chap. V. Book I.) Rankine states that in some experiments made lately on well-covered cylinders, the energy thus wasted was greater than that given out by the engine. He continues:—"The remedy for this cause of loss is to prevent that spontaneous liquefaction of the steam during its expansive working, in which the process just described originates; and that is done either by enclosing the cylinder in a jacket, or casing, supplied with hot steam from the boiler, or by superheating the steam before its admission into the cylinder, or by both these means combined. The steam is thus kept in a nearly dry state, so as to be a bad conductor of heat, and the moisture which it contains, though sufficient to lubricate the piston, is not allowed to increase to such an extent as to carry away any appreciable quantity of heat from the metal of the cylinder and piston to the condensers."*

* Rankine's "Memoir of John Elder," p. 20.

After speaking of Watt's introduction of the steam-jacket, he says:—"The use of the steam-jacket was retained in a few special kinds of engines, such as the Cornish pumping-engines; and in them the economy properly due to high rates of expansion of the steam was realized; but in almost all other engines, and certainly in marine engines, the jacket was abandoned, with this result—that little or no practical advantage was found to result from expansive working when the steam was expanded to more than about double, or two and a half times its original volume; and this became a received maxim amongst engineers, and especially amongst marine engineers, for its truth in the case of unjacketed cylinders was established by practical experience, as well as by experiments made for the purpose of testing it." *

In Art. 86 we pointed out the importance of allowing the first expansion to occur in a small cylinder, the loss from heat given up by the steam-jacket during exhaust being thus lessened as well as the heat lost in radiation; for even if the second or large cylinder is also jacketed, its temperature is much lower than the jacket of the small cylinder. But there is another great benefit arising from an increase in the number of cylinders which was first pointed out by *Elder*. Namely, that the pressures on the crank shaft may be made so to balance that friction may be almost destroyed. Elder, in an unpublished paper read in the United Service Institution in 1866, showed that it was advantageous to employ a compound engine with *two* cylinders when expanding steam to *five* times its volume, and one with three cylinders for successive expansion to *ten* times the volume.

More credit is due to Elder, perhaps, than to any other practical engineer for the improvements in marine-engineering economy. In 1856 he applied

* Page 21.

engines with four cylinders in which the above principle was carried out, and the engines designed by him in 1858 are nearly perfect in this way.

With the introduction of high pressures and quick speeds in marine engines came the necessity for the *balancing* of the moving masses, the principles of which were discussed in Art. 214. Readers must be referred to larger treatises and actual engines for information as to how these principles are carried out in practice.

In American river-engines, very high-pressure steam is used, say of 100 to 150 lb. per square inch. These engines would therefore be much more economical (Art. 130) than ordinary steamboat engines if the workmanship were the same in both cases. Uncouth, quick-moving beam-engines may often be seen, with trussed timber beams and framing.

CHAPTER VII.

PADDLE-WHEEL ENGINES.

275. **The Side-Lever Engine.**—This is a modification of the ordinary stationary beam-engine. There are two *side-levers*, or sway beams, J K, of cast or wrought-iron (Fig. 74), one on each side of the engine, placed in this position for compactness, and that the centre of gravity of the vessel may be as low as possible. The end of the piston-rod D is guided by means of the parallel motion E F D. Parallel motions in marine engines are gradually giving way to slides. The *cylinder cross-head* at D is bolted and cuttered to the piston-rod at its middle point, and communicates motion at its extremities to the ends of the side-

levers by two rods D K. The main centre passes through the condenser at H, being supported on pedestals outside. The ends J of the side-levers are connected by the *cross-tail*, which communicates motion to the crank-pin by means of the connecting-rod J L.

The long D slide-valve is much in use in engines of this kind. It is now thought right for all marine engines to be provided with the link-motion.

The condenser P H Q is represented as being part of

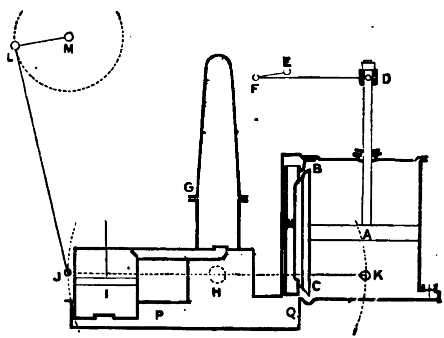

FIG. 74.

the sole-plate in the roughly-drawn figure, but it is always thought better to make it a separate casting.

The framing is now seldom of cast iron, being more usually formed of wrought-iron stays. The framing ought not to be attached to the sides of the ship.

276. **Direct-action Engines.**—It is probable that for paddle-wheels the steeple engine with two or more piston-rods and the ordinary oscillating engine will supersede all others. The following engines are direct-acting.

277. *Gorgon Engine*—Direct-action engines dispense with the side-levers, the piston-rod being directly attached to the connecting-rod. The chief peculiarity of the Gorgon engine is its parallel motion, which

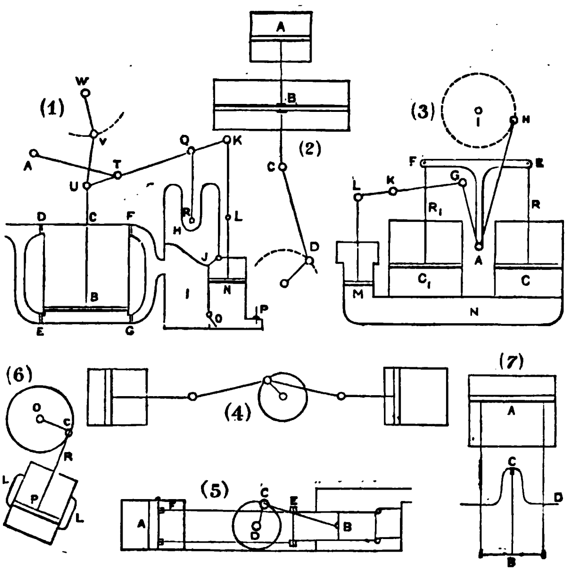

FIG. 75.

is shown in UTQR (1, Fig. 75). QR is a rocking beam, moving about R as a centre and constraining the point Q to have a small motion in a circular arc. The motion of the beam UK is constrained by the two

radius bars A T and R Q; so that the parallel motion is that shown in Fig. 31, Art. 109. K L is a connecting-rod working the air-pump rod, for which guides are provided at L. Sometimes, instead of producing U Q, the radius-bar T A is produced in the direction of A to work the air-pump.

The parts are very compactly arranged, but the crank shaft is at a great height even when a short connecting-rod is used, and this is the principal objection to all engines of the Gorgon class.

278. **The Steeple Engine.**—Here the guides for the piston-rods are greatly elevated. the connecting-rod descending to the crank, which is placed immediately over the cylinder cover. It is usual to provide two piston-rods one on either side of the crank-shaft. This form of engine is much employed in river steamers both in England and America.

279. *Maudslay's Two-cylinder Engine.*—This was designed to give a long connecting-rod without rendering great height necessary. Two pistons are raised and lowered simultaneously, their rods carrying the ends of a T piece F E A (3, Fig. 75). A moves in guides attached to the cylinder, and a connecting-rod A H works the crank H I. G K L is a lever working the air-pump M. N is the condenser.

There was a cylindric slide-valve (Art. 80) worked by an eccentric to admit steam at the same instant to both cylinders.

There is much radiation of heat from the double-cylinder engine, there is little compactness, and the condenser is too long to enable the air-pump to work well in a heavy sea.

280. *Messrs. Miller and Ravenhill*, adopting the Gorgon form, constructed a very compact engine, in which the condenser casting connected two cylinders, and included two air-pumps. These air-pumps were inclined and were both worked from a crank on the in-

termediate shaft. Inclined pumps are found to work more easily than those which are vertical.

This firm afterwards adopted the principle of Penn's oscillating engines, now to be described; retaining, however, their own inclined air-pumps.

281. **The Oscillating Engine.**—Of all direct-action engines for the paddle-wheel, this is perhaps the most satisfactory in compactness and facility of being repaired. 6, Fig. 75 gives an idea of how it differs in construction from ordinary engines.

No connecting-rod is needed, the piston-rod, which has a brass head with a cutter and screws (Art. 112), is directly attached to the crank-pin, and the cylinder has power to move about its centre of gravity on trunnions.

Steam passes through the belts L L surrounding the cylinders, one-half of a belt being always in connection with the boiler, the other half with the condenser. The pipes to the boiler and condenser, respectively, are turned on the outsides of the ends and allow trunnions fixed to the cylinders to slide easily round them in such a way that there is no hindrance to the flow of steam. The trunnions are now usually cast separate and bolted to the cylinder; they rest on plummer blocks, which are bolted to the framing. The large stuffing-boxes required for the trunnions have metallic packing backed with hemp.

The wear at these bearings is inconsiderable, and it is found that the presence of steam prevents any great rise in temperature.

Ribs from the belt, particularly near the trunnions, are required to strengthen the cylinder, which is otherwise made as strong and rigid as possible to prevent strains.

The slide-valves are usually of the three-ported or locomotive kind (Art. 90), and are placed at L and L.

Originally, there was only one valve-casing on the

cylinder, and it was balanced by an equal mass of metal placed on the opposite side; but the plan now adopted has many advantages. As the casing oscillates with the cylinder, the valve-rod cannot be directly connected with the eccentric; it has guides attached to the cylinder, and the further extremity of the lever which gives it motion is a brass block which slides easily in a circular slot.

The centre of the circular arcs of the slot is on the centre line of the trunnions, so that the lever moves easily with the cylinder.

The moveable slot-frame is attached to the end of the eccentric-rod, and its extremities are guided by vertical columns which they half embrace, so that the valve gets the same motion from the eccentric as if the cylinder were stationary. This moveable slot-frame may be worked independently of the eccentric whenever necessary.

There are usually two air-pumps placed and worked like those of the engines of Messrs. Miller and Ravenhill, or when a crank is thought objectionable by means of a large broad eccentric. The condenser is usually cast on the lower framing, and extends beneath the trunnions.

The upper and lower cast-iron framings are bound together by means of strong wrought-iron rods.

It was at first thought that the cylinders of oscillating engines would rapidly wear oval, but it is found that this effect is produced so slowly as to be almost immaterial. The evil is lessened by providing the piston with a projecting rim round the edges to increase the bearing surface. There is much tendency in the stuffing-box to wear oval, and hence it is made of great depth. The piston-rod is made of steel, and is much larger than usual.

CHAPTER VIII.

SCREW-ENGINES.

282. **With Gearing.**—When screws were first introduced it was supposed impossible to get engines to work them directly, no smooth-working fast engines having yet been constructed. Hence, large spur-wheels were made to move pinions on the screw-shaft, and as the gearing worked very smoothly and the momentum of the large spur-wheel was useful in heavy seas in preventing "racing," geared engines were found to be efficient for driving the screw-propeller, and they may still be employed with advantage in merchant steamers. However, the smaller size and compactness of direct-action engines, and for vessels of war, the possibility of all their parts being kept far below the water-line, have led to the general adoption of the latter.

283. **Direct-action Engines.**—Cranks on the screw-shaft are worked by the engine directly, and hence the piston-velocity is much greater here than in any other form of marine engine. As a rule, the pistons are on opposite sides of the shaft, as also are the air-pumps, &c. The piston-rod for steadiness and to prevent wear is usually produced so as to pass through a stuffing-box in the bottom of the cylinder. These engines are nearly all provided with the link-motion.

Direct-action engines in confined places have many defects, the greatest of which is the use of a short connecting-rod.

The success with which *compact* powerful engines have been constructed can only be recognized after an examination of the working drawings given in larger treatises, such as *Burgh's Marine Engineering* and its

Appendix. Students are directed to make careful studies of such drawings, and of real engines. We proceed to describe the prominent features of different types of direct-action engines.

284. *Seaward's Engines.*—In 4, Fig. 75 two horizontal cylinders on opposite sides of the shaft work one crank, one centre line passing through the axes of the cylinders and the centre of the shaft. Altogether, there are four cylinders, and two vertical air-pumps of short stroke are worked from two eccentrics. Many vessels of the royal navy are provided with engines of this construction.

285. *Messrs. Boulton and Watt* employ four oscillating cylinders, lying on their sides, instead of those of last example, the piston-rods being directly attached to cranks on the screw-shaft. There is a link-motion provided for the valves, which are of the ordinary locomotive kind ; but these are supplemented by separate expansion slide-valves worked from elliptical cams on the shaft.

The air-pumps are inclined and are wrought by means of separate cranks or eccentrics.

286. **Penn's Trunk Engines.**—These owe their great success mainly to the excellent adjustment of their details and to their good workmanship. One of the two

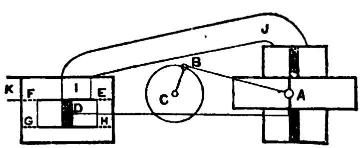

FIG. 76.

engines is shown in Fig. 76, where A is a cylinder of great diameter. The piston is cast on the trunk, like a collar on a shaft. The connecting-rod A B is attached

to the piston and trunk at A by means of a strong pin. The trunk is of sufficient diameter to allow the connecting-rod to have its proper motion. The horizontal double-acting air-pump D (*see* Art. 100) is worked from a small piston-rod which is independent of the trunk. This air-pump is placed within the condenser and has the same length of stroke as the piston. The feed and bilge pumps are worked in the same manner, having in every case the stroke of the piston. J is the eduction-pipe sloping downwards to the condenser and providing partial condensation by its great surface.

The great exposure of the heated surface of the trunk at every stroke is the principal defect in these engines.

287. *Napier's Trunk Air-pump Engine* is very efficient. It has almost all the compactness of the trunk-engine without its loss of heat in radiation.

Two piston-rods F E (5 Fig. 75) are attached to lugs on a trunk E B, forming the plunger of the air-pump. Water leaves the condenser by one of the valves seen at the end of the pump, during the back stroke, and passes up into the hot well by the other valve during the forward stroke.

The trunk has an important part to play in rigidly connecting the piston-rods and connecting-rod B C.

288. Modifications of the *Horizontal Steeple-engine* have been constructed by Holm, Bourne, Maudslay, and others. The ends of the piston-rods are attached to an ordinary slide, and thus by means of backward-moving connecting-rods turn the crank-shaft which is placed just in front of the cylinder-cover. The air-pumps are double-acting and horizontal, and are worked from projections on the end of the cross-head.

Maudslay's engine is shown in 7, Fig. 75.

Blyth constructed a good engine, in which the piston-rods of two upright oscillating cylinders caught two

corners of an upright triangular frame, the third corner of which was attached to the crank-pin.

Thomson inverted his cylinders, the centre line through the piston-rod, the slides and the crank-shaft being vertical, and in this manner obtained an engine which was well suited for screw vessels of the merchant service. It is a form much followed by engineers. (*See* Frontispiece.)

Carlsun's engine was of a very good form. The cylinders were inclined downwards towards the shaft from opposite sides, thus greatly economizing space. In the existing examples of this engine the details have not been well adjusted.

Ericcson in the *Princeton,* U.S.N., employs hinged pistons, which swing through angles of about 120°, the two piston-shafts having cranks whose motions of partial revolution are converted into the revolution of a crank on the screw-shaft by means of connecting-rods.

The construction of *Humphrey's Compound Engines* may be understood from 2, Fig. 75. The piston in the high-pressure cylinder A and that of the low-pressure cylinder B are rigidly connected by means of one long piston-rod, which gives motion to the crank D E. This engine is fitted to ships of war by being laid upon its side, by the piston-rod A B not being continued, and by introducing a trunk from B instead of the arrangement B C D.

289. Most modern engines belong to one or other of the above classes. In matters of detail, particularly in the arrangement of pumps and condensers, great diversity occurs, the tendency being towards compactness, with easy access to all the different parts for examination. Most of the compound engines described in Art. 86 are at present working in steam-vessels, and a number of examples may be examined from working drawings in the *Appendix* to *Burgh's Marine Engineering.* (*See* Frontispiece.)

CHAPTER IX.

GENERAL DETAILS OF MARINE ENGINES.

290. **Cylinder.**—The bottom is more frequently cast with the cylinder in marine than in land engines. In these cases a hole is cast in the bottom, that a boring-bar may pass through, and a plug is let in. Very large cylinders are often cast open at the bottom with a turned flange, and are bolted to the sole-plate by means of a great number of iron or copper bolts. A turned projecting part of the sole-plate fits into the bottom. In oscillating engines the bottom is always cast on the cylinder.

291. After the important differences mentioned in the last chapter between the engines of different makers, nothing has so much effect on the shape of engines as the position and nature of the *pumps and condensers.* Many different compact arrangements of air pumps with jet and surface condensers are illustrated in *Burgh's* Book on *Condensation.* In *jet* condensers a cock is provided for allowing bilge-water to be injected when necessary. The injection-pipe has no rose head *inside* the condenser because of the liability to choke up with small bits of refuse.

In *surface* condensers the brass or copper tubes are straight; they are sometimes *drawn,* but as a small flaw may be produced in the drawing, and as small flaws are apt to become great, some makers prefer *soldering.* The tubes may be $\frac{5}{8}''$ or more in diameter, and are usually about 6 feet long. They are fixed in the tube-plates by means of ferules of compressed pine which expand when wetted, or by means of brass screws and linen or India-rubber washers. The pumps for the condensed and the condensing water are of

many kinds, the latter being sometimes a centrifugal pump worked by a donkey-engine. In a promising arrangement, due to American engineers, a vacuum is maintained both inside and outside the tubes, which are thus subjected to but little bursting pressure, and may be made very thin, and therefore much more efficient for conducting heat from the condensing steam to the water. Bourne recommends the use of 1 square foot of tube-surface per cubic foot of water evaporated by the boiler per hour in ordinary cases. He further states that oil ought to be used for lubricating the piston instead of tallow, and that it ought to be fed continuously without waste, else the tubes of the condenser will get choked up, or the steam will carry verdigris produced by the oil in the tubes into the boiler to the hurt of the iron. As this will always occur to a slight extent, he recommends as a remedy that the boiler be fed with some salt water occasionally.

292 In marine engines we are prevented employing those efficient arrangements for expansion employed on stationary engines, tappets and conical valves, because of the pitching of the vessel, and the slide arrangements of Art. 96 become of importance. As a general rule, the faces of the slide valves used are large, and special arrangements are made to lessen the friction. It is usual to have a hollow at the back of the valve, surrounded by a ring with a plane face, capable of sliding steam tight on the casing, the hollow being connected with the condenser. If the ring and recess are on the casing, and the back of the valve is quite plane, the brass or iron ring may be backed by india-rubber or wrought iron rings, and may be tightened on the valve by means of screws outside the casing.

293. *The Eccentric* is always in two pieces. Unless when a link-motion is used, the eccentric is loose and has a back-balance and catches.

It is usual to place the eccentric-rod in the notch by means of a pulley, another pulley at the same instant disconnecting the starting-lever, as it is inconvenient for the long lever to be swaying about when the eccentric is in gear. Instead of a starting-lever it is now more usual to employ a wheel with spokes like the steering-wheel of a ship.

The links in large engines are moved by means of a separate steam-cylinder and piston. In the engines of the *Ulster* the links are moved from this piston, the rod of which is hollow and screwed inside, fitting a long screw, which turns with a fly-wheel when the piston moves, and thus injurious shocks are prevented.*

294. The most important shafts of screw-engines and the small shafts of paddle-engines are now usually made of steel. It is said that the *outside* only of a piston-rod ought to be converted into steel. The bearings of quick moving shafts are in length usually 3 to $3\frac{1}{2}$ times their diameters.

CHAPTER X.

MARINE BOILERS.

295. **Flue Boilers.**—Figs. 44 and 45 give an idea of the general configuration of a flue boiler. It will be seen that as many eddies as possible have been produced in flues so limited in length. Flue boilers are too large and too heavy, yet many engineers prefer them to the tubular kind. They are much thought of in the Cunard line.

296. **Tubular Boilers.**—Figs. 46 and 47 give an idea of the general configuration of a tubular boiler. In

* See Bourne on the Screw-propeller

front of the tube-plate at D there is a door which enables the tubes to be cleaned. As a rule, large marine tubular boilers have this or a somewhat similar construction.

Napier's boiler, which is very efficient and is much used for Clyde steamers, has small vertical water tubes in the flues, and resembles some vertical land boilers.

In very efficient boilers of *Rowan*, of Glasgow, the flues are vertical and are thin rectangular spaces, well stayed by means of cross-pipes of rectangular section.

In the *Great Eastern* we see how sets of boilers may be placed back to back. These boilers are fixed at the ends, the furnaces of every two have a common bridge, and the heated gases pass through two sets of tubes from a common combustion chamber to a common chimney.

In some *American river steamers*, in which wood is burnt, there is a flue from the furnace, and six large tubes convey smoke to the chimney from the back of boiler.

In some cases many small vertical water tubes are placed in a descending flue behind the furnace; the bottom of the boiler is much below the furnace, and the flue passes just above the bottom to the chimney which is at the end remote from the grate.

In many vessels of the royal navy, which have low boilers, the tubes are on a level with the furnace and are liable to get choked up with cinders, &c. The smoke-box doors, which allow the tubes to be cleaned and require an empty space in front of them, are at the end remote from the furnace. The opinion is growing among engineers, that there is no great necessity for low boilers, that an armour-plate will protect as well as a few feet of water, and that the above disadvantages, together with the liability to break of horizontal cylinders, and, among other sources of annoyance, the compelled shortness of connecting-rods

render necessary a return to high boilers and to upright steeple-engines in ships of war.

Tubes are generally of iron, six to seven feet in length, fixed by being driven firmly into the tube-plates and slightly riveted into a countersink at the ends. Tubes of brass need ferules. It is found that when leakage is prevented there is but little galvanic action with brass tubes. Tube-plates have wrought-iron stays with two nuts, one on each side of the tube-plate. Sometimes the *tubes* are screwed at the ends to act as stays. The staying of marine boilers differs so much in boilers which differ even slightly in construction, and depends so much on circumstances, that students must here be referred for information to the plates of large works on marine engines.

297. *Furnace-bars* must not be longer than six feet. Fire-brick *bridges* seem preferable to all others. *Flues* of different boilers ought, if possible, to be kept distinct, as it is convenient in large vessels to clean out a different boiler every watch.

Chimneys.—Funnel-plates are usually 9 feet long and $\frac{3}{8}$ inch thick. Two hoops for *stays* ought to be provided. The *waste steam-pipe* ought to be as high or higher than the chimney, that the plates may not be hurt. The *damper* is usually at the end of the boiler-flue. A pipe leading overboard ought to be provided for the escape of the condensed steam from the safety-valve.

298. **Corrosion of Marine Boilers.**—Metals are apt to corrode when subjected to the intermittent action of moisture and dryness; as inside a boiler just at the surface of the water where the plates are seldom covered or uncovered with water for a great length of time. Also, dripping water from the deck on the top of a boiler, bilge water at the bottom, and the water sometimes thrown into the ashpit are great sources of corrosion.

The top is now usually covered with felt and sheet-lead, the boiler is bedded properly with mastic cement, which is afterwards beaten into corners by means of wooden tools, and the ashpits are provided with spare moveable plates. It is to be remembered that no copper bolts from the sides of the ship or from the wooden platform ought to touch the bottom boiler-plates.

The unexplained corrosion which sometimes occurs in the steam-chest is not so easily remedied. It appears to arise from the heating effect of the chimney-flue, which passes through the chest, and, indeed, it seems to occur whenever there is superheated steam in the chest. Bourne recommends casing the flue with fire-brick at this place, or else allowing part of the feed-water to pass through a space between the steam-chest and the chimney, or passing the flue away from the chest altogether. I think it may be a feasible and efficient plan to superheat the steam in a strong casing surrounding the flue, as well as by means of tubes, in such a way that there will be no superheated steam in the chest.

299. The formation of **Scale** will be prevented by judicious blowing-off. All sea-water has approximately the same composition, although there is less salt in some water than in others. Hence we must always be prepared for the deposition of chloride of magnesium and carbonate of calcium. The introduction of sal-ammoniac is not sufficient to prevent incrustation, there must also be a proper amount of *blowing-off*. The introduction of sal-ammoniac merely causes the deposition to be made in the form of a fine mud, which is even worse in its non-conductivity than the ordinary scale. The use of pure feed-water and surface condensation is the only plan which will completely prevent the formation of scale.

300. Sea-water varies in density from 1·026 to 1·029.

It contains about ·03 of its weight of salt and at the pressure of one atmosphere (14·7 lb. per square inch) will boil at 100°·66 C. When the amount of salt is doubled the temperature of boiling is 101°·32, and, in general, the temperature of boiling gets higher by 0·66 of a degree centigrade for every additional 0·03 of the weight of the water of salt, so that at its point of saturation, namely, when it contains 0·36 of its weight of salt, the temperature of boiling is 107°·9 C. If the *pressure is greater* than that of the atmosphere (as it always is in boilers), it will be found that the temperature corresponding to the pressure (Art. 32) of the steam is *always* increased by 0·66 of a degree centigrade for every additional 0·03 of its weight in salt added to the water.

301. Salt also increases the density of water. If the density of pure water is 1, that of water containing ·03 of its weight of salt is 1·029, and there is an addition of ·029 to the density for every additional ·03 of its weight in salt added to the water.

Partly from the less solubility of the calcium carbonate and magnesium chloride, water must be blown off when it has about 0·09 or 0·10 of its weight of salt.

302. **Salinometers.**—These are contrivances for determining the degree of saltness of the water in a boiler by means of the temperature of boiling, or else by means of the density of the water.

(1.) **By the Thermometer.**—Water is taken from the boiler, and is boiled in the engine-room at the pressure of one atmosphere (14·7 lb. per square inch). If the temperature of boiling is above 102 C.—that is, if it contains more than ·09 of its weight of salt—it ought to be blown off.

For accurate determinations when the atmospheric pressure in the engine-room is different from 14·7 lb. per square inch, the barometer ought to be observed, and from the Table (Art. 32) the corresponding temperature of boiling of pure

water may be found. If this differs by more than two pegrees from the actual temperature, the water ought to be blown off.

(2.) **By the Hydrometer.**—Hydrometers are instruments for determining the densities of liquids. The hydrometer here used for determining the density of the water is a hollow ball of glass or metal with a long stem. The ball is weighted below, so that the instrument swims upright in the water; the depth to which it sinks determining the density. It is always graduated for some particular temperature, so that we must have the water at this temperature before the hydrometer can be used. As a rule, hydrometers are graduated for the temperatures 93°·3 C., or 17°·8 C. It is usual to put the mark W or O for the level of ordinary seawater, and the indications of other densities are denoted by arbitrary numbers. In the third and fourth columns of the following table are given the indications on the ordinary hydrometers used by engineers.

Indications on Thermometers and Hydrometers for different degrees of saltness of water.

Saltness.	Boiling point at pressure of 1 atmo-phere.	Indications of the two Hydrometers.	
		93°·3	17°·8
·03	100°·66 C.	W or O	12
·06	101·32	6	20
·09	101·98	12	28
·12	102·64	18	36
·15	103·30	24	44
·18	103·96	36	50

303. It may be shown that if the brine blown off be supposed to contain n times as much salt as the feed-water, we must blow off a quantity equal in weight to $\frac{1}{n}$th that of the feed-water.

For the brine must contain all the salt of the feed-water, and n lb. of feed contain as much salt as 1 lb. of brine; hence, 1 lb. of brine is drawn off for n lb. of feed.

We see then that for the water in a boiler to contain at all times 0·09 of its weight of salt, we ought to *blow off one-third* as much water as we feed with.

The brine is taken away by means of blow-off cocks, brine-pumps, blowing off at the surface, or sediment collectors.

304. **Blow-off Cocks.**—After letting the level of water gradually rise in the boiler, a cock is opened to the sea and closed when a sufficient quantity of water has been blown out. It is better to have a small cock continually letting water escape, the amount of opening being governed by the indications of a salinometer. It is necessary at all times to have a salinometer in sight in the engine-room.

Every boiler has its own blow-out cock, and besides one common cock is provided for all. These cocks ought to be quite close to the boilers.

305. **Brine-pumps and Valves.**—Brine-pumps are so constructed that they extract an amount of brine equal to a certain fraction of the feed, one-third or one-fourth. In large vessels they are often worked by donkey engines. Seaward connects the feed and brine valves, in such a way that they are open at the same time, their areas being say as 3 to 1. When these are used the ordinary blow-off cocks are also opened occasionally.

306. **Surface Blow-off.**—Particles of sediment tend to come to the surface of water during ebullition;

cocks opened just below the surface will receive such particles.

307. **Sediment Collectors.**—An inverted hollow cone whose base rises above the water level, has openings near the base, through which water enters, the sediment depositing in the still water and falling into a tube from which it is led overboard.

308. **Kingston's Valves.**—Pipes opening below the surface of the outside water are provided with valves of the shape shown in Fig. 77, which may be closed by means of a hand-wheel and a spindle with a long screw.

CHAPTER XI.

PRACTICAL WORKING OF MARINE ENGINES.

309. **After arrival in Port.**—The engine must be *blown through* and water removed from the cylinder, condenser, air-pump, and hot-well. If an accident has happened to the blow-through valve, steam may be admitted to one side or other of the piston by means of the slide-valve and subsequently to the condenser. Horizontal cylinders ought afterwards to be carefully dried.

Before blowing through it may be necessary to determine if the *piston* is *steam-tight* by admitting steam on the bottom-side, and opening the grease-cocks of the cover. Holding-down bolts and stays are carefully examined and adjusted before the steam is let down.

The boiler must be kept perfectly dry inside, or else well filled with water to prevent oxidation, so that when a short stay in port is intended it is not well to blow off on arrival. The boiled water contains no air and will produce few bad effects. The level of the water ought to be raised from time to time, to prevent oxidation at the surface.

If *scale* is to be removed, it will be necessary to empty the

boiler. The blow-off cocks ought to be examined afterwards, as the pipes are apt to get stopped up with pieces of scale washed in by the water. For this reason boilers when emptied should be pumped quite dry and the blow-off pipe ought to be provided with a coarse grating.

The engine must be *cleaned down* while warm. The packings from the piston, &c., must be removed that there may be no oxidation when a long stay is intended; for a short stay melted tallow must be thrown into the grease-cups round the piston-rods to catch entering dirt, and ought to be removed when the steam is being got up again. It will sometimes be necessary to scald certain lubricators with hot water before getting up steam. Bright parts of the engine are

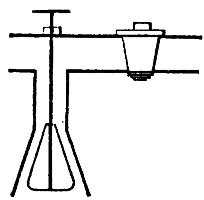

FIG. 77.

usually coated with a mixture of melted tallow and white lead before proceeding to sea.

The slide-valve may be taken out and its worn faces adjusted by filing and scraping.

Steam ought to be got up from time to time while in harbour during a short stay *to give a few turns* to the engine and to keep it in working order. If this is not allowed, the engine ought to be *turned by hand*. In screw-engines a worm and wheel are often provided for turning the screw-shaft from time to time.

310. **Getting up Steam.**—The Kingston's valves and gauge-cocks may be opened to fill the boiler.

When the gauge-cocks are kept closed very little water will enter, as the internal pressure will soon balance that due to difference in levels outside the ship and in the boilers.

Sometimes, in vessels of light draught, hand-pumps must be employed after the water has reached a certain height, the Kingston's valves being closed.

Whilst the water is entering, the fires are lighted by covering the grate with coals, closing the ashpit-door and setting on fire a mass of wood, oakum, &c., on the dead-plate. When the ignited wood is scattered over the grate the coals burn rapidly, but it is not for some time that the ashpit-door is opened.

When the flues are too cold the mixed gases cannot attain the temperature required for combustion, and hence it is advisable to merely cover the flues, or tubes, at first with water, admitting more gradually as the flues get sufficiently hot. For rapidity of draught it will sometimes be necessary to open all the hatchways of the engine-room.

It will be well to keep the safety-valve closed, as the heated air of the boiler will prove useful in warming the engine; the throttle-valve ought to be tried from time to time as it sometimes expands and sticks fast.

The engine is examined for obstructions to the moving parts, such as blocks of wood left carelessly about in the air- and feed-pumps, &c. Ice may be removed from the air and feed-pumps by blowing through for a short time, and from the bilge-pumps by heating from the outside.

The *funnel-stays* must be slackened, and afterwards made tight as the different parts attain constant temperatures.

If the *injection orifice is stopped up* with mud or weeds, the snifting-valve and waste water-pipe must be closed, so that steam may be blown through the injection-pipe. This will in general give sufficient relief.

Steam must now be blown through so as to clear the engine of air and water, which escape by the snifting-valve, or when there is no snifting-valve, through the air-pump bucket, the valves, the hot well, and waste water-pipe overboard.

A bucket of water thrown once or twice on the *snifting-valve* when it draws air will sometimes get rid of the piece of wood or other obstruction which gives annoyance.

To *start the engines* for the first time we must know with what kind of slide-valve they are fitted, and as it is well to know exactly how to move the starting-gear at once whenever

necessary, the engineer ought to make himself acquainted with the relative motions of the crank and of the starting lever—that is, at some suitable time he must observe the effect of moving the starting-lever into various positions.

It is sometimes necessary to disconnect the valves from the eccentric, *and to work them by hand for a few strokes* that steam may be admitted during the whole stroke, and that water may have an opportunity of escaping by the slide. The throttle and injection-valves may be only partially open at first until the ship gathers way.

Circumstances must decide whether or not to start before the steam is well up; if about to steam through an intricate passage or out of a crowded harbour, it is necessary to wait.

A boiler is apt to prime at first, as great quantities of steam of low pressure are taken away rapidly and there is violent ebullition, perhaps aggravated by the presence of mud.

The steam-valves may be partially closed and the fires put back in this case. Priming is often got rid of by introducing tallow, but the plan is not to be recommended.

311. **When Under Steam.**—The fires may be *put back slightly* by closing the ashpit-doors and the dampers, or by opening the smoke-box doors. Fires are *banked up* when the engine stops and is not likely to start for about half an hour. This is effected by putting the fires back against the bridge and covering with wet small coal and ashes. During this time the boiler-pressure may be lowered. Twenty or thirty minutes may be necessary to get the steam well up after allowing its pressure to become as low as that of the atmosphere with a banked-up fire.

Clinkers must not be allowed to remain in the fires, which ought to be *cleaned out* at regular intervals and left clean by each set of men at the end of their watch. Water must never be *thrown* on the *hot ashes in the ashpit. Hot ashes* and *burning coal from the funnel* indicate great waste. *Flame from the funnel* indicates an insufficient supply of air. *Insufficient draught* may be remedied by introducing a small steam jet into the funnel.

After having allowed brine to accumulate it may be necessary to slacken speed a little on blowing out.

When proceeding slowly on a voyage it will be found more

economical to use *slow combustion in all* than to employ a few boilers only. If the blow-out cock sticks fast and there are no brine-pumps, the valves of the boiler hand-pump may be reversed and employed to let the water escape.

The water-level in the boilers ought to be high enough to prevent the tubes or flues getting uncovered from the ordinary rolling of the vessel.

The glass gauge is liable to get choked up.

When the cylinder is jacketed care must be taken that the jacket does not fill with condensed water; hence the jacket-cock ought to be always slightly open.

Variations in the proper temperature for the condenser will depend on the depth of the foot-valve below the surface of the sea, on the temperature of the injection-water and on the exhaust or back-pressure of steam. Generally, the best temperature for the condenser is about 22 degrees above the temperature of the injection-water. In a heavy sea it may be necessary to raise the temperature, as a high back-pressure will prevent "racing," and, besides, during the sudden checks to motion the injection-water being limited in supply will be unable to fill the condenser.

Injection-water may, on a leak occurring, be taken from the bilge. It is necessary to use precautions against the entrance of chips of wood or other refuse which might hurt the pump-valves. On an emergency, condensing engines may be made to work like non-condensing.

The Kingston's valves fitted to the sea-cocks ought to be keyed open. When there is a leak in the bottom of the condenser a temporary receptacle for water is made round the bottom, and this water is allowed to enter through the leak. The injection-cock may be partially closed when this occurs.

Leaks in the engine valve-chest may be caulked for the time with spun yarn or hemp soaked in white or red lead, or by means of a wedge of wood covered with white lead driven in tightly.

The expansion gear ought to be thrown out when the engine stops, because that at certain parts of the stroke there is no admission for steam on one side or other of the piston.

The engineer must examine the coal bunkers from time to time to guard against spontaneous combustion.

312. **During Action.**—Much depends on the coolness and decision of the engineer when the machinery gets deranged. He ought to be prepared with a proper course of action for any emergency, no matter how sudden. Hence it is his duty to make a careful study of his engine whatever its construction may be.

Before action the disconnecting gear and gear for injecting from the bilge ought to be examined with special care, and the donkey-engine and hand-pumps put in readiness to extinguish fires.

The state of the furnace-fires must be regulated with reference to the probable duties of the vessel, whether these are chasing, avoiding, &c., or else involve slow movements. During an action it is of the greatest consequence to let the boiler-pressure be kept near that of the atmosphere.

Paddle-wheels which have suffered considerable damage may still be found capable of propelling the vessel. A hole in the funnel merely produces a diminution in the draught. After the action, this hole may be covered with a thin iron plate held in its place by means of springs. There may be a little danger from flames when a spent shot knocks the funnel down, but even in this case the principal inconvenience will be to the people on deck from smoke. A graver accident is that of the bending of the steam-pipe and consequent stoppage of steam when the vessel is at rest. When the vessel is in motion the cylinder will consume the steam. In the other case, it will be well to blow through the hot-well and waste water-pipe overboard. The grease-cocks in the cylinder cover may be opened, steam being admitted above the piston. With this method of getting rid of the steam the engine is kept in working order. In some cases it may be possible to let the vessel steam ahead and astern for short distances.

Steam ought to be up in all the boilers before the action, some of the fires being kept banked up.

When a shot strikes the water-space of a boiler, the safety-valve is opened, steam for the engine is taken from this boiler alone, and the hot water is allowed to run into the bilge. If the steam-space is struck, the pressure of the steam is immediately reduced, if not already the same as that of the atmo-

sphere, when the hole may be covered with a piece of wood made steam-tight by means of fearnaught and white lead, and kept in its place by means of a stay to the ship's side.

After the action the shell may be temporarily repaired by means of properly curved plates. These ought to have been in readiness provided with bolts and cross-bars for placing inside the boiler. With the help of fearnaught and white or red lead, steam-tight joints may be made.

Damaged tubes ought to be plugged up with long plugs from one or both ends. If several are damaged the fires may be removed altogether.

Holes in steam and feed-pipes may be covered with a curved sheet of copper or iron tied round with rope-yarn.

During the repairing of the feed-pipe it may be possible to feed one boiler from the other; if not, and there is no separate feed-pump, close the stop-cock, blow out all the steam, and let water enter from the sea by the blow-off cocks and Kingston's valves at intervals.

In many cases it may be found necessary to work the engine without condensation, when the waste-water pipe is injured for instance.

If one engine is disabled a float may be taken from that part of the paddle-wheel which is lowest when the remaining engine is at a dead point. Sails may be necessary to enable the engine to get up sufficient speed to get past its dead points.

After an accident, if the vessel has to be run ashore, let the boilers be filled, and let water enter the bilge that she may take the ground soon and may afterwards be easily lightened.

With a broken cylinder cover the engine may be made single-acting by plugging the upper port. The plug may be held by two stays to the opposite side of the cylinder.

MISCELLANEOUS EXERCISES.

1. A tube whose section is circular is replaced by four tubes of the same total volume; show that the heating surface is thereby doubled.

2. The brine-pump of a boiler is choked. How is the brine to be got rid of, the steam-gauge indicating 4 lbs. per square inch above that of the atmosphere, and the upper surface of the water being 2 feet below the level of the sea?

Ans. By blowing out. The excess of pressure is 1 lb. per square inch.

3. Find the H.P. of a locomotive which draws a train of T tons at the rate of v miles per hour, the friction being at the rate of f lbs. per tons.

$$Ans.\ \frac{88vTf}{33000}.$$

4. In a locomotive boiler there are 156 tubes, each 2 inches in diameter and 127 inches long. What is the amount of the tube heating-surface?

Ans. 864,464 square feet.

5. A pair of double-cylinder engines (the two cylinders being the same) is substituted for single-cylinder engines of 78 inches diameter; if the total area of the piston and length of the stroke be the same in both cases, compare the surfaces of the cylinders exposed to the rubbing of the pistons.

$$Ans.\ \sqrt{2} : 1.$$

6. A screw makes 73 revolutions per minute. Find its pitch if the slip is $\frac{1}{4}$ and the speed of the ship is 14 knots.

Ans. 25·9 feet.

7. A safety-valve, 4 inches in diameter, is constructed so that each pound of additional pressure per square inch on the valve corresponds to 1 lb. pressure on the spring. What are the relative distances of the spring and of the valve from the fulcrum? If the spring requires 10 lbs. to extend it one inch, what additional pressure will lift the valve $\frac{1}{20}$ of an inch after the valve is set.

Ans. 2 : 25 ; ·497 lbs.

8. Find the angle of a screw propeller, diameter 16 feet, pitch 20 feet.

Ans. $21° 41'$.

9. If feed-water at 37·8 contains $\frac{1}{30}$ of its weight of salt, and if the boiler-water at 120° is made to contain $\frac{1}{10}$ of its weight of salt before it is blown off, what percentage of the total heat given to the boiler is wasted by blowing off?

Ans. 6·5 per cent.

10. A boiler safety-valve is loaded to 20 lbs. per square inch. The vessel heels over 25° when steam makes its escape. Find the steam pressure.

Ans. $20 \times \cos 25°$ or 18·12 lb.

11. A pair of engines of 850 indicated H.P. which give a speed of 9 knots to a vessel are replaced by others which give 11 knots. What is the indicated H.P. of the new engines?

Ans. 1552 nearly.

12. A ship has 4 boilers. With 2 boilers working at the ordinary pressure the speed is 7 knots; what is the speed with 3 boilers?

Ans. 8·013 knots.

13. There is a bar of length L and cross section A. It is gradually elongated by an amount l; if its modulus of elasticity be E, show that the work expended on its elongation is

$$\frac{l^2}{2L} KE.$$

14. In what time would an engine of 15 H.P. (indicated)

empty a shaft full of water, the diameter of the shaft being 8 feet, and the depth 1,200 feet? (The whole weight of water may be supposed to be lifted from half the depth of the well.)

Ans. 76 hours.

14.* If the engine of last example has a duty of 30 millions, determine the amount of coal consumed in emptying the shaft.

Ans. 75·4 bushels.

15. The pressure of air being 14·7058 lbs. per square inch, find the height of a column of mercury whose base is one square inch, to represent the pressure of the atmosphere, the specific gravity of mercury being 13·596.

Ans. 29·905 inches.

16. What must be the diameter of a single-acting pump with a stroke of 39 inches and which is $\frac{3}{5}$ full at each stroke, to lift 300 tons of salt water per hour if the engine makes 45 revolutions per minute, and if the specific gravity of salt-water is 1·0267?

Ans. 19·12 inches.

17. Find the volume of water lifted in 5 hours by a brine-pump $3\frac{3}{4}$ inches diameter, 12 inches stroke, 10 strokes per minute; the pump being $\frac{2}{3}$ full every stroke.

Ans. 153·4 cubic feet.

18. Find the normal H.P. of a steam-engine, 25 inch cylinder, 3 feet stroke, 55 revolutions per minute.

Ans. 34·36.

19. In a tubular boiler there are 144 tubes, each $2\frac{1}{4}$ inches in diameter and 10 feet long. Find the amount of heating surface, if the heating surface around the fire-box be 40 square feet.

Ans. 888·232 square feet.

20. Given a normal indicator diagram. Show what change will take place in its form if the injection-water be shut off; or, secondly, if the steam be throttled; thirdly, if the steam passage be opened too late and the exhaust passage too soon. Draw the indicator diagram which would be taken from the

cylinder of a single-acting air pump. How would the diagram tell you when the water was being delivered?

What kind of diagram would be obtained from the two sides of a piston if the slide-rod were shortened?

What are the faults of the engines whose diagrams are shown in Fig. 78?

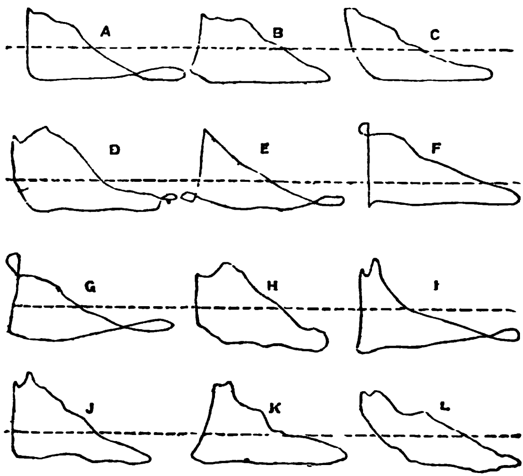

FIG. 78.

21. The stroke of the piston of a locomotive is 24 inches, the diameter of the driving-wheel is 8 feet. What is the mean velocity of the piston when the engine is running at 40 miles per hour?

Ans. 560 feet per minute.

22. Show how to find the work done by the piston in turning the crank in a direct-action engine. What force applied

to the extremity of the crank, at right angles to it in all positions, will do the same work as a mean piston-rod pressure of 4 tons during a revolution.

Ans. 2,546 tons.

23. In a double-beat valve, the internal diameters of the two seats are 5 and $3\frac{1}{4}$ inches, the weight of the valve is 68 lbs.; what pressure per square inch, or head of water, will cause it to lift?

Ans. 5·99 lbs. or 15·57 feet.

24. The travel of a slide is to be increased from 13 to 15 inches. What alteration must be made in the length of the lever by means of which the valve is worked from the eccentric, its original length being 12 inches?

Ans. Shortened $1\frac{5}{8}$ inches.

25. Diameter of screw 16 feet, angle 21° 10′; find the pitch.

Ans. 19·46 feet.

26. If the heat given to a feed-water heater is really saved; find the saving effected in 24 hours by increasing the temperature of 85 gallons of feed-water per hour from 15° to 85°. A gallon of pure water weighs 10 lbs.

Ans. 1428000 units of heat.

27. Find the pitch of a screw-propeller which makes 72 revolutions per minute; the speed of the vessel being 10 knots, and slip being 12 per cent.

Ans. 16 feet nearly.

28. Length of crank 2 feet, connecting-rod 6 feet. Find the angle between the connecting-rod and crank when the piston is at the middle of the cylinder. The engine is direct acting.

Ans. 80° 25′.

29. A bar of wrought-iron 100 feet long, 2 square inches in section, has its temperature raised from 0° to 100°. How many units of work has the heat done in lengthening the bar?

Ans. 3875 units.

30. Water enters a boiler from a tank. The difference in height between the surfaces being 35 feet. What is the steam-pressure when the water is just prevented from entering?

Ans. 15·18 lbs. above the atmosphere.

31. The lever of a safety-valve is 16 inches long, the spindle acts at 4 inches from the fulcrum; the diameter of the valve is 4 inches. Find the weight which may be placed at the end of the lever to allow steam to escape at the pressure of 45 lbs.

Ans. 94·25 lbs.

32. Prove this rule

$$\text{Normal H.P.} = \frac{\text{diam.}^2 \times \text{speed of piston}}{6000}.$$

33. Find the pressure necessary to keep a Kingston's valve closed when 20 feet below the water surface, and in connexion with a vacuum of $13\frac{1}{4}$ lbs. per sq. inch.

Ans. 22·14 lbs.

34. Find the load on an air-pump bucket which is 16 feet below the surface of the water outside the ship, the pressure of the atmosphere being 14·75 lbs. and the pressure in the condenser being 2·5 lbs.?

A cubic foot of sea-water weighs 64 lbs.

Ans. 19·36 lbs. per square inch.

35. If the speed is 10 knots when the H.P. is 560, what will be the speed when the H.P. is 800, supposing the new speed to be "moderate" for the vessel?

Ans. 11·2 knots.

36. A steamer is 500 miles from port which bears due W. and has coal for only 300 miles. Shall she get within *steaming* distance by sailing on a N.W. by W. course?

Ans. Yes. Within 277·8 miles.

37. The positive lap at one port of a locomotive slide valve is $3\frac{1}{4}$ inches, and at the other port is $3\frac{3}{8}$ inches; the greatest admission at each port is $3\frac{3}{8}$ inches: find the travel of the slide.

The whole throw to each side beyond its middle position is evidently lap + admission; hence the whole travel $= 3\frac{1}{4} + 3\frac{3}{8} + 3\frac{3}{8} + 3\frac{3}{8}$, or $13\frac{3}{8}$ inches.

38. A slide has $3\frac{1}{4}$ inches positive lap and $\frac{1}{4}$ inch negative lap at a certain port. How far must the slide travel from the point at which expansion begins before the port opens to exhaust?

Ans. $3\frac{1}{2}$ inches.

39. Two well-shaped vessels of the same size start on a voyage of 700 and 850 nautical miles respectively. One has a speed of 9 knots, the other with less expansion in the cylinder has a speed of 10 knots, and consumes 200 tons of coal on the voyage; what is the consumption of the first? By Art. 237 the consumption varies as the length of the voyage $\times$ square of speed. The *answer* is 133·4 tons.

APPENDIX.

SCIENCE AND ART DEPARTMENT.

EXAMINATION IN STEAM, 1872.

*Seven Questions may be attempted in any one Paper.**

ELEMENTARY.

1. DEFINE the *temperature* of a body. What two natural phenomena, taking place at definite temperatures, have been employed to determine two points of reference in the scale of a thermometer? How is a thermometer graduated according to Fahrenheit's scale?

2. Distinguish between a gas and a liquid. What is the relation between the pressure and volume of a gas at a given temperature? A certain quantity of steam at 40 lbs. pressure is admitted into a cylinder, and then expands to four times its original bulk; what is its final pressure? What do you understand by steam at 40 lbs. pressure?

3. Describe Newcomen's atmospheric engine, and point out its defects. Why was this engine not suited for driving machinery?

* The other directions have been left out.

4. Certain valves commonly used in a steam-engine are called "slide-valves." Sketch some valve of this kind, and explain its action. By what operation is the face of such a valve made truly plane?

5. Enumerate the principal parts of a condensing beam-engine.

6. Describe the governor of a steam-engine. In what way does it act upon the throttle-valve? The position of the balls being known for a certain speed of rotation, what change will occur if the speed be doubled?

7. Sketch a throttle-valve; this is an "equilibrium," or "balanced" valve. Why is it so called? Explain any other form of equilibrium-valve in common use.

8. What gauge should you employ for measuring the pressure of the steam in the boiler of an engine? It will be necessary to explain the principle on which the gauge is constructed.

9. The steam-cylinder of a trunk engine is 60 inches in diameter, and the trunk is 24 inches in diameter; what is the diameter of an ordinary steam-cylinder of the same effective area?

10. Describe, with a sketch, some form of marine tubular boiler. How is the water-gauge made and fitted?

11. In what way is the water in a marine boiler relieved from an excess of salt? Explain the operation of testing for the quantity of salt dissolved in the water. Estimate the number of ounces of salt dissolved in a gallon of water which boils at 214° F.

12. Describe briefly the arrangement of a side-lever engine. Where are the foot and delivery valves respectively placed? What is the object of the blow-through valve? How would you start the engine?

13. What change must be made in the position of the eccentric pulley upon the shaft in reversing an engine, and for what reason?

14. Describe Kingston's valve, and sketch it in section.

15. What are the principal features of a locomotive boiler? Describe the construction of the fire-box. In what way is the fire-box protected from the danger of collapsing under the pressure to which its surfaces are exposed?

16. During what part of the stroke is an ordinary locomotive slide-valve pressed by the steam most strongly against the face of the ports? Make your meaning clear by a diagram.

17. How is the piston of a locomotive engine packed so as to be steam-tight?

18. A train is running at 40 miles per hour, the driving-wheel is 8 feet in diameter, and the throw of the crank is 2 feet. What is the average linear velocity of the steam-piston?

19. Explain the use of two eccentrics in reversing a locomotive engine.

19.* Describe any system of switches by which a train can be shifted from the main line into a siding. For what purpose are guard-rails placed by the side of a rail? Point out the manner in which a guard-rail acts.

ADVANCED.

20. Define the unit of heat; how many units of heat will be required in order to boil away and convert into steam one pound of water at 212°? Estimate the approximate bulk of the steam produced.

21. Mention experiments which show that water is a very bad conductor of heat. Describe the process by which a mass of water becomes heated throughout. State any circumstances which may cause the boiling point of water to rise above 212, and give some reason for the fact in each case.

22. Steam enters a cylinder at 15 lbs. pressure, it is cut off at $\frac{1}{3}$rd of the stroke, the resistance from uncondensed vapour averaging 2 lbs. per inch; find the mean effective steam-pressure (given $\log_e 3 = 1{\cdot}099$).

23. Describe some form of indicator. Sketch the indicator diagrams which you would expect to obtain from the top and bottom of the cylinder of a *single-acting* condensing-engine.

24. Explain the geometrical principle of Watt's parallel motion. Do you know any arrangement which gives an exact parallel motion?

25. Sketch a steam-valve and the ports, giving (1) lap to the steam side only, (2) lap to both the steam and exhaust sides. What is the effect of adding lap to the exhaust side of the valve?

26. The co-efficients of linear expansion of wrought-iron and brass are given in a table as being ·00118 and ·00189 respectively between 32° and 212°. If a tube of brass and a tube of iron be each 11 feet long at 32°, what will be the difference in their lengths when heated to 340°?

27. Explain Bourdon's pressure-gauge.

28. What is the principle of the gridiron valve? How is this principle applied in constructing the slide-valves of large marine engines?

29. Describe Penn's trunk engine. Sketch in section the steam-cylinder and condenser, with the pump and valves.

30. The pitch of a screw-propeller is 24 feet, it makes 60 revolutions per minute, and the slip is 23 per cent. Find the speed of the ship in knots per hour.

31. What is superheated steam? Describe some superheating apparatus. Is the pressure of the steam increased by superheating it? What change does the extra heating produce?

32. Explain the method of reversing an engine when fitted with a single eccentric.

33. Sketch the feed-pump and valves connected with it as fitted to the boiler of a marine engine. What is the position of the escape or overflow valve? When is an air vessel desirable?

34. A locomotive slide-valve is 10½ inches long and 17 inches wide; when at one end of the stroke its outer edge overlaps the face of the port by ⅛-inch, what is the total steam-pressure on the back of the valve? Does this pressure increase or diminish as the valve moves forward so as to close the steam-port?

35. Give some explanation of Giffard's injector, and sketch it in section.

36. The driving-wheel of a locomotive engine is 8 feet in diameter, and the engine is running at 40 miles per hour; what are the linear velocities of the highest, middle, and lowest points of the wheel at any moment?

37. Describe the construction of the fire-box of a locomotive boiler; the general form of the fire-box being rectangular, in what way is each flat surface strengthened so as to support the pressure of the steam?

38. Describe some friction brake, and calculate the power you gain by any combination of levers and screw which you may employ.

39. Explain Stephenson's link motion.

HONOURS.

40. When a portion of a gas is enclosed in a cylinder and compressed by a piston, show that the work done by the piston against the gas is equal to the diminution in volume of the gas multiplied by the average hydrostatic pressure.

41. How is an "indicator" diagram traced out? Prove that the area of the curve represents the amount of work done by the steam during each reciprocation of the piston.

42. Explain Giffard's injector.

43. Prove that it is economical to work steam expansively. Describe the arrangement of a beam-engine with high and low pressure cylinders. What valves should you employ for distributing the steam?

44. Find the time of rotation of a conical pendulum. Prove that when the balls are suspended so as to move in a parabola, they will retain any position indifferently so long as the speed of rotation remains constant.

45. Describe and explain the method of reversing an engine by double eccentric and link motion.

46. When a condensing-engine is fitted with independent steam and exhaust valves, what form of valve should you employ, and how should you arrange for expansive working?

47. Explain the valve motion of a marine oscillating engine.

48. Calculate the increase in the expenditure of steam for a given amount of clearance. What is the most advantageous adjustment for cushioning?

49. In a direct-acting engine find the angle the crank has moved through when the piston is at a given distance from

one end of the stroke. If the crank has moved through the same angle at the top and bottom of the stroke, compare the respective movements of the piston.

50. When a steam-cylinder is not furnished with a steam-jacket, it is considered that a certain quantity of steam will pass through the cylinder without doing any work. Account for this result, and explain the advantage of superheating steam.

51. Find the area of the blade of a screw-propeller of given dimensions.

In the following Table we give empirical rules governing the relative proportions of the different parts of engines. These have been obtained from comparison of engines by the best makers. They are quite unimportant when compared with rules deduced by the student himself from good engines constructed to meet requirements similar to those needed in his particular case. When time allows it is always best to *calculate* the dimensions of the different parts of the engine, employing the principles of Applied Mechanics, gross mistakes being prevented by comparison with a similar engine by a good maker.

A = area of piston. d = diam. of piston, in inches.

	Condensing Land Beam Engine.	Non-Condensing Land Engine.	Screw Engines.	Oscillating Engine.
Stroke	$d \times 2\frac{1}{4}$ to 2	$d \times 2$	Average $d \div 2$	$d \times {}^{\cdot}8$ to 1
Piston-rod. Diameter	$d \div 10$	$d \div 6$ to 7	$d \div 9$ to 11	$\left\{ d \div {}^{\cdot}8 \right.$ to 10, use 10 for large Cyls.
Cylinder. Thickness	$\frac{d+5}{40}$	$\frac{d+5}{20}$	$\frac{d+5}{50}$	$\frac{d+5}{50}$
Connecting-rod. Length	Stroke × 3	Stroke × 2, as a minimum	Average stroke × 2.5	..
Connecting-rod. Greatest Cross Section ..	Cast $A \div 18$ Wt. $A \div 21$	$A \div 21$, taper $\frac{1}{8}''$ for 1′	$A \div 18$, taper $\frac{1}{8}''$ for 1′	..
Crank-pin. Diameter	$d \times {}^{\cdot}14$	$d \times {}^{\cdot}25$	Diameter of Piston-rod × 2	Diameter of Piston-rod × 1·2
Crank-pin. Length	$d \times {}^{\cdot}21$	$d \times .25$	Diameter of Pin × ·75	Diameter of Pin × 1·5
Crank-shaft. Diameter of bearing	$d \times {}^{\cdot}25$	Diameter of Piston-rod × 2	Diameter of Piston-rod × 2	Diameter of Piston-rod × 1·8 to 1·6
Length of Bosses, wrought-iron Crank ..	For all Engines, Diameter of Shaft or Pin.			

	Condensing Land Beam Engine.	Non-Condensing Land Engine.	Screw Engines.	Oscillating Engine.
Fly-wheel. Diameter	Stroke × 3 to 2·5	Stroke × 3 to 2	..	..
Fly-wheel. Rim, in cwts.	Nominal H.P. × 2 to 3	Nominal H.P. × 2 to 1·5	..	..
Fly-wheel. Depth of Rim	Dia. of Wheel ÷ 16 to 20	Dia. of Wheel ÷ 8 to 9	..	..
	FOR ALL ENGINES.			
Vol. of Air-pump	Single acting, Vol. of Cyl. ÷ 6 to 7. Double acting, Vol. of Cyl. ÷ 10 to 12.			
Vol. of Feed-pump. Cubic inches	Vol. of Cyl. in cubic feet × 3 to 4·5 times the number of cubic inches of water required for 1 cubic ft. of steam.			
Vol. of Condenser	With one Cyl., Vol. of Cyl. ÷ 6 to 7. With two Cyls., Vol. of one Cyl. ÷ 4.			
Diameter Air-pump Rod	Diameter of Pump ÷ 6 to 8.			
Diameter Feed-pump Rod	Diameter of Plunger + 2 to 3.			

TABLE OF DIMENSIONS OF SIDE-LEVER MARINE ENGINES OF SEAWARD & Co.

Nominal H.P.	Diam. of Cylinder.	Diam. of Air-pump.	Stroke in feet.	Diam. of Paddle-wheel.	Main Shaft.	Piston-rod.	Air-pump Rod (Copper).	Length of Beam.	Beam Gudgeon	Diam. Crank-pin.	Length Crank-pin.
10	20	12	2	9	4	2	1·25	6	3	2·5	2·75
20	27	17	2·5	11	5·75	3	1·75	8	5	3·5	3·75
40	36·5	20	3	13	7·5	3·5	2·25	10	6	4·5	4·35
60	43	24	4	17	9	4	2·75	11·5	7	5·5	6
80	48	27	4·5	19	10	4·5	3·25	13	8	6·5	7
100	52·5	..	5	21	11	5	3·75	16	9	7·37	7·75
120	57·5	..	5·5	23	12·5	5·5	4·25	17·5	9·75	8	8·5

TABLE OF SPECIFIC GRAVITIES.

	Specific Gravity.		Specific Gravity.
Brass, Wire	8·544	Steel, Hammered	7·840
„ Cast	8·399	Tin	7·291
Bronze	8·222	Zinc	7·190
Copper, Wrought	8·915	Basalt	2·650
„ Wire..	8·878	Brick	1·557 to 2·168
„ Cast	8·788	Brickwork in Cement ..	1·680
Gun Metal	8·784	„ Mortar ..	1·568
Iron, Wrought	7·758	Granite	2·7
„ Cast	7·093	Beech	·777
Lead, Cast	11·446	Boxwood	1·32 to ·91
„ Sheet	11·407	Oak	·934 to ·756
„ Wiredrawn	11·317	Pine	·66 to ·46

LOGS FOR STEAMER.

I.—Made at the end of each Watch by Engineer in charge.

Dates.	Hours of Watch.	Boilers.					Condensers.			Temperatures.					Expansion.	Revolutions per minute. Average.	Knots per Hour. Average.			Remarks.
		Number in use.	Number of Fires.	Coals burnt during Watch.	Average Pressure of Steam.	Saltness by Scale of Salinometer.	External Barometer.	Engine Barometer, Starboard.	Engine Barometer, Larboard.	Hot Well, Starboard.	Hot Well, Larboard.	Engine-room.	Coal-boxes. Maximum.	Sea.	Cut off.		Wheels or Screw.	Ship, by Patent Log.	Slip.	
		Tons..	..	..	= {Coals burnt during preceding 12 hours.										Averages =					

II.—Made from (I.) by the First Engineer every 12 hours and given to the Captain.

DATES.	Knots per hour, Wheels or Screw.	Ship by Patent Log.	Distance run in preceding 12 hours.	Number of Hours in which Sails were set.	Number of Hours under Steam.	Distance run from beginning of Voyage.	Indicated H.P.	STORES.			
								Coals.		Tallow used in preceding 12 hours.	Oil used in preceding 12 hours.
								Burnt in preceding 12 hours.	Remaining in Ship.		

TABLE OF HYPERBOLIC LOGARITHMS.

The Hyperbolic or Napierian Logarithm of a number may be obtained from the ordinary Logarithm of the number by multiplying by 2·30258505.

Num.	Hyp. Log.	Num.	Hyp. Log.	Num.	Hyp. Log.	Num.	Hyp. Log.	Num.	Hyp. Log.
1·05	·049	3·05	1·115	5·05	1·619	7·05	1·953	9·05	2·203
1·1	·095	3·1	1·131	5·1	1·629	7·1	1·960	9·1	2·208
1·15	·140	3·15	1·147	5·15	1·639	7·15	1·967	9·15	2·214
1·2	·182	3·2	1·163	5·2	1·649	7·2	1·974	9·2	2·219
1·25	·223	3·25	1·179	5·25	1·658	7·25	1·981	9·25	2·225
1·3	·262	3·3	1·194	5·3	1·668	7·3	1·988	9·3	2·230
1·35	·300	3·35	1·209	5·35	1·677	7·35	1·995	9·35	2·235
1·4	·336	3·4	1·224	5·4	1·686	7·4	2·001	9·4	2·241
1·45	·372	3·45	1·238	5·45	1·696	7·45	2·008	9·45	2·246
1·5	·405	3·5	1·253	5·5	1·705	7·5	2·015	9·5	2·251
1·55	·438	3·55	1·267	5·55	1·714	7·55	2·022	9·55	2·257
1·6	·470	3·6	1·281	5·6	1·723	7·6	2·028	9·6	2·262
1·65	·500	3·65	1·295	5·65	1·732	7·65	2·035	9·65	2·267
1·7	·531	3·7	1·308	5·7	1·740	7·7	2·041	9·7	2·272
1·75	·560	3·75	1·322	5·75	1·749	7·75	2·048	9·75	2·277
1·8	·588	3·8	1·335	5·8	1·758	7·8	2·054	9·8	2·282
1·85	·615	3·85	1·348	5·85	1·766	7·85	2·061	9·85	2·287
1·9	·642	3·9	1·361	5·9	1·775	7·9	2·067	9·9	2·293
1·95	·668	3·95	1·374	5·95	1·783	7·95	2·073	9·95	2·298
2·0	·693	4·0	1·386	6·0	1·792	8·0	2·079	10·0	2·303
2·05	·718	4·05	1·399	6·05	1·800	8·05	2·086	15	2·708
2·1	·742	4·1	1·411	6·1	1·808	8·1	2·092	20	2·996
2·15	·765	4·15	1·423	6·15	1·816	8·15	2·098	25	3·219
2·2	·788	4·2	1·435	6·2	1·824	8·2	2·104	30	3·401
2·25	·811	4·25	1·447	6·25	1·833	8·25	2·110	35	3·555
2·3	·833	4·3	1·459	6·3	1·841	8·3	2·116	40	3·689
2·35	·854	4·35	1·470	6·35	1·848	8·35	2·122	45	3·807
2·4	·875	4·4	1·482	6·4	1·856	8·4	2·128	50	3·912
2·45	·896	4·45	1·493	6·45	1·864	8·45	2·134	55	4·007
2·5	·916	4·5	1·504	6·5	1·872	8·5	2·140	60	4·094
2·55	·936	4·55	1·515	6·55	1·879	8·55	2·146	65	4·174
2·6	·956	4·6	1·526	6·6	1·887	8·6	2·152	70	4·248
2·65	·975	4·65	1·537	6·65	1·895	8·65	2·158	75	4·317
2·7	·993	4·7	1·548	6·7	1·902	8·7	2·163	80	4·382
2·75	1·012	4·75	1·558	6·75	1·910	8·75	2·169	85	4·443
2·8	1·030	4·8	1·569	6·8	1·917	8·8	2·175	90	4·500
2·85	1·047	4·85	1·579	6·85	1·924	8·85	2·180	95	4·5[illegible]4
2·9	1·065	4·9	1·589	6·9	1·931	8·9	2·186	100	4·605
2·95	1·082	4·95	1·599	6·95	1·939	8·95	2·192	1,000	6·908
3·0	1·099	5·0	1·609	7·0	1·946	9·0	2·197	10,000	9·210

TABLE OF DIAMETERS AND AREAS OF CIRCLES.

Diam. in Inches.	Area.	Diam. in Inches.	Area.	Diam. in Inches.	Area.	Diam. in Inches.	Area.
		5	**19·635**	**10**	**78·540**	**15**	**176·715**
1/8	·0122	1/8	20·629	1/8	80·515	1/8	179·672
1/4	·0490	1/4	21·647	1/4	82·516	1/4	182·654
3/8	·1104	3/8	22·690	3/8	84·540	3/8	185·661
1/2	·1963	1/2	23·758	1/2	86·590	1/2	188·692
5/8	·3068	5/8	24·850	5/8	88·664	5/8	191·748
3/4	·4417	3/4	25·967	3/4	90·762	3/4	194·828
7/8	·6013	7/8	27·108	7/8	92·885	7/8	197·933
1	**·7854**	**6**	**28·274**	**11**	**95·033**	**16**	**201·062**
1/8	·9940	1/8	29·464	1/8	97·205	1/8	204·216
1/4	1·2271	1/4	30·679	1/4	99·402	1/4	207·394
3/8	1·4848	3/8	31·919	3/8	101·623	3/8	210·597
1/2	1·7671	1/2	33·183	1/2	103·869	1/2	213·825
5/8	2·0739	5/8	34·471	5/8	106·139	5/8	217·077
3/4	2·4052	3/4	35·784	3/4	108·434	3/4	220·353
7/8	2·7611	7/8	37·122	7/8	110·753	7/8	223·654
2	**3·1416**	**7**	**38·484**	**12**	**113·097**	**17**	**226·980**
1/8	3·5465	1/8	39·871	1/8	115·466	1/8	230·330
1/4	3·9760	1/4	41·282	1/4	117·859	1/4	233·705
3/8	4·4302	3/8	42·718	3/8	120·276	3/8	237·104
1/2	4·9087	1/2	44·178	1/2	122·718	1/2	240·528
5/8	5·4119	5/8	45·663	5/8	125·184	5/8	243·977
3/4	5·9395	3/4	47·173	3/4	127·676	3/4	247·450
7/8	6·4918	7/8	48·707	7/8	130·192	7/8	250·947
3	**7·0686**	**8**	**50·265**	**13**	**132·732**	**18**	**254·469**
1/8	7·6699	1/8	51·848	1/8	135·297	1/8	258·016
1/4	8·2957	1/4	53·456	1/4	137·886	1/4	261·587
3/8	8·9462	3/8	55·088	3/8	140·500	3/8	265·182
1/2	9·6211	1/2	56·745	1/2	143·139	1/2	268·803
5/8	10·320	5/8	58·426	5/8	145·802	5/8	272·447
3/4	11·044	3/4	60·132	3/4	148·489	3/4	276·117
7/8	11·793	7/8	61·862	7/8	151·201	7/8	279·811
4	**12·566**	**9**	**63·617**	**14**	**153·938**	**19**	**283·529**
1/8	13·364	1/8	65·396	1/8	156·609	1/8	287·272
1/4	14·186	1/4	67·200	1/4	159·485	1/4	291·039
3/8	15·033	3/8	69·029	3/8	162·295	3/8	294·831
1/2	15·904	1/2	70·882	1/2	165·130	1/2	298·648
5/8	16·800	5/8	72·759	5/8	167·989	5/8	302·489
3/4	17·720	3/4	74·662	3/4	170·873	3/4	306·355
7/8	18·665	7/8	76·588	7/8	173·782	7/8	310·245

TABLE OF DIAMETERS AND AREAS OF CIRCLES—*continued.*

Diam. in Inches.	Area.	Diam. in Inches.	Area.	Diam. in Inches.	Area.	Diam. in Inches.	Area.
20	**314·160**	**25**	**490·875**	**30**	**706·860**	**35**	**962·11**
1/8	318·099	1/8	495·796	1/8	712·762	1/8	968·99
1/4	322·063	1/4	500·741	1/4	718·690	1/4	975·90
3/8	326·051	3/8	505·711	3/8	724·641	3/8	982·84
1/2	330·064	1/2	510·706	1/2	730·618	1/2	989·80
5/8	334·101	5/8	515·725	5/8	736·619	5/8	996·78
3/4	338·163	3/4	520·769	3/4	742·644	3/4	1003·7
7/8	342·250	7/8	525·837	7/8	748·694	7/8	1010·8
21	**346·361**	**26**	**530·930**	**31**	**754·769**	**36**	**1017·87**
1/8	350·497	1/8	536·047	1/8	760·868	1/8	1024·95
1/4	354·657	1/4	541·189	1/4	766·992	1/4	1032·06
3/8	358·841	3/8	546·356	3/8	773·140	3/8	1039·19
1/2	363·051	1/2	551·547	1/2	779·313	1/2	1046·39
5/8	367·284	5/8	556·762	5/8	785·510	5/8	1053·52
3/4	371·543	3/4	562·002	3/4	791·732	3/4	1060·73
7/8	375·826	7/8	567·267	7/8	797·978	7/8	1067·95
22	**380·133**	**27**	**572·556**	**32**	**804·249**	**37**	**1075·21**
1/8	384·465	1/8	577·870	1/8	810·545	1/8	1082·48
1/4	388·822	1/4	583·208	1/4	816·865	1/4	1089·79
3/8	393·203	3/8	588·571	3/8	823·209	3/8	1097·11
1/2	397·608	1/2	593·958	1/2	829·578	1/2	1104·46
5/8	402·038	5/8	599·370	5/8	835·972	5/8	1111·84
3/4	406·493	3/4	604·807	3/4	842·390	3/4	1119·24
7/8	410·972	7/8	610·268	7/8	848·833	7/8	1126·66
23	**415·476**	**28**	**615·753**	**33**	**855·30**	**38**	**1134·11**
1/8	420·004	1/8	621·263	1/8	861·79	1/8	1141·59
1/4	424·557	1/4	626·798	1/4	868·30	1/4	1149·08
3/8	429·135	3/8	632·357	3/8	874·84	3/8	1156·61
1/2	433·731	1/2	637·941	1/2	881·41	1/2	1164·15
5/8	438·363	5/8	643·594	5/8	888·00	5/8	1171·73
3/4	443·014	3/4	649·182	3/4	894·61	3/4	1179·32
7/8	447·699	7/8	654·839	7/8	901·25	7/8	1186·94
24	**452·390**	**29**	**660·521**	**34**	**907·92**	**39**	**1194·59**
1/8	457·115	1/8	666·227	1/8	914·61	1/8	1202·26
1/4	461·864	1/4	671·958	1/4	921·32	1/4	1209·95
3/8	466·638	3/8	677·714	3/8	928·06	3/8	1217·67
1/2	471·436	1/2	683·494	1/2	934·82	1/2	1225·42
5/8	476·259	5/8	689·298	5/8	941·60	5/8	1233·18
3/4	481·106	3/4	695·128	3/4	948·41	3/4	1240·98
7/8	485·978	7/8	700·981	7/8	955·25	7/8	1248·79

TABLE OF DIAMETERS AND AREAS OF CIRCLES—*continued.*

Diam. in Inches.	Area.	Diam. in Inches.	Area.	Diam. in Inches.	Area.	Diam. in Inches.	Area.
40	**1256·64**	**45**	**1590·43**	**50**	**1963·50**	**60**	**2827·44**
1/8	1264·50	1/8	1599·28	1/4	1983·18	1/4	2851·05
1/4	1272·39	1/4	1608·15	1/2	2002·96	1/2	2874·76
3/8	1280·31	3/8	1617·04	3/4	2022·84	3/4	2898·56
1/2	1288·25	1/2	1625·97	**51**	**2042·82**	**61**	**2922·47**
5/8	1296·21	5/8	1634·92	1/4	2062·90	1/4	2946·47
3/4	1304·20	3/4	1643·89	1/2	2083·07	1/2	2970·57
7/8	1312·21	7/8	1652·88	3/4	2103·35	3/4	2994·77
41	**1320·25**	**46**	**1661·90**	**52**	**2123·72**	**62**	**3019·07**
1/8	1328·32	1/8	1670·95	1/4	2144·19	1/4	3043·47
1/4	1336·40	1/4	1680·01	1/2	2164·75	1/2	3067·96
3/8	1344·51	3/8	1689·10	3/4	2185·42	3/4	3092·56
1/2	1352·65	1/2	1698·23	**53**	**2206·18**	**63**	**3117·25**
5/8	1360·81	5/8	1707·37	1/4	2227·05	1/4	3142·04
3/4	1369·00	3/4	1716·54	1/2	2248·01	1/2	3166·92
7/8	1377·21	7/8	1725·73	3/4	2269·06	3/4	3191·91
42	**1385·44**	**47**	**1734·94**	**54**	**2290·22**	**64**	**3216·99**
1/8	1393·70	1/8	1744·18	1/4	2311·48	1/4	3242·17
1/4	1401·98	1/4	1753·45	1/2	2332·83	1/2	3267·46
3/8	1410·29	3/8	1762·73	3/4	2354·28	3/4	3292·83
1/2	1418·62	1/2	1772·05	**55**	**2375·83**	**65**	**3318·31**
5/8	1426·98	5/8	1781·39	1/4	2397·48	1/4	3343·88
3/4	1435·36	3/4	1790·76	1/2	2419·22	1/2	3369·56
7/8	1443·77	7/8	1800·14	3/4	2441·07	3/4	3395·33
43	**1452·20**	**48**	**1809·56**	**56**	**2463·01**	**66**	**3421·20**
1/8	1460·65	1/8	1818·99	1/4	2485·05	1/4	3447·16
1/4	1469·13	1/4	1828·46	1/2	2507·19	1/2	3473·23
3/8	1477·63	3/8	1837·93	3/4	2529·42	3/4	3499·39
1/2	1486·17	1/2	1847·45	**57**	**2551·76**	**67**	**3525·66**
5/8	1494·72	5/8	1856·99	1/4	2574·19	1/4	3552·01
3/4	1503·30	3/4	1868·55	1/2	2596·72	1/2	3578·47
7/8	1511·90	7/8	1876·13	3/4	2619·35	3/4	3605·03
44	**1520·53**	**49**	**1885·74**	**58**	**2642·08**	**68**	**3631·68**
1/8	1529·18	1/8	1895·37	1/4	2664·91	1/4	3658·44
1/4	1537·86	1/4	1905·03	1/2	2687·83	1/2	3685·29
3/8	1546·55	3/8	1914·70	3/4	2710·85	3/4	3712·24
1/2	1555·28	1/2	1924·42	**59**	**2733·97**	**69**	**3739·28**
5/8	1564·03	5/8	1934·15	1/4	2757·19	1/4	3766·43
3/4	1572·81	3/4	1943·91	1/2	2780·51	1/2	3793·67
7/8	1581·61	7/8	1953·69	3/4	2803·92	3/4	3821·02

TABLE OF DIAMETERS AND AREAS OF CIRCLES—*continued.*

Diam. in Inches.	Area.	Diam. in Inches.	Area.	Diam. in Inches.	Area.	Diam. in Inches.	Area.
70	**3848·46**	**78**	**4778·37**	**86**	**5808·81**	**94**	**6939·79**
¼	3875·99	¼	4809·05	¼	5842·63	¼	6976·75
½	3903·63	½	4839·83	½	5876·55	½	7013·81
¾	3931·36	¾	4870·70	¾	5910·57	¾	7050·97
71	**3959·20**	**79**	**4901·68**	**87**	**5944·69**	**95**	**7088·23**
¼	3987·13	¼	4932·75	¼	5978·90	¼	7125·58
½	4015·16	½	4963·92	½	6013·21	½	7163·04
¾	4043·28	¾	4995·19	¾	6047·62	¾	7200·59
72	**4071·51**	**80**	**5026·56**	**88**	**6082·13**	**96**	**7238·24**
¼	4099·83	¼	5058·02	¼	6116·74	¼	7275·99
½	4128·25	½	5089·58	½	6151·44	½	7313·84
¾	4156·77	¾	5121·24	¾	6186·25	¾	7351·78
73	**4185·39**	**81**	**5153·00**	**89**	**6221·15**	**97**	**7389·82**
¼	4214·11	¼	5184·86	¼	6256·15	¼	7427·96
½	4242·92	½	5216·82	½	6291·25	½	7466·20
¾	4271·83	¾	5248·87	¾	6326·44	¾	7504·54
74	**4300·85**	**82**	**5281·02**	**90**	**6361·74**	**98**	**7542·98**
¼	4329·95	¼	5313·27	¼	6397·13	¼	7581·51
½	4359·16	½	5345·62	½	6432·62	½	7620·14
¾	4388·47	¾	5378·07	¾	6468·21	¾	7658·87
75	**4417·87**	**83**	**5410·62**	**91**	**6503·89**	**99**	**7697·70**
¼	4447·37	¼	5443·26	¼	6539·68	¼	7736·62
½	4476·97	½	5476·00	½	6573·56	½	7775·65
¾	4506·67	¾	5508·84	¾	6611·54	¾	7814·77
76	**4536·47**	**84**	**5541·78**	**92**	**6647·62**	**100**	**7854·00**
¼	4566·36	¼	5574·81	¼	6683·80		
½	4596·35	½	5607·95	½	6720·07		
¾	4626·44	¾	5641·18	¾	6756·45		
77	**4656·63**	**85**	**5674·51**	**93**	**6792·92**		
¼	4666·92	¼	5707·94	¼	6829·49		
½	4717·30	½	5741·47	½	6866·16		
¾	4747·79	¾	5775·09	¾	6902·92		

INDEX.

INDEX.

A.

B.

5757

THE END.

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